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They Don't Know It All

WILL IRWIN, writing in *Liberty* his impressions of the great men that he has known, all leaders in one field or another, says:

As I review the character of the eminent men I have known, I find one common factor. They are always teachable.

The big fellow knows that he does not know it all and is ever on the lookout for information.

That is how he gets to know more than the average man and to be a big fellow.

He is ever ready to profit by another's experience or thinking and seeks the association and conversation or reads the writings of those who are working and thinking in the lines in which he is interested.

He is afflicted with no contempt for "booklearning".

No man, however wide his sphere of activity or however great his capacity for observation and absorption, knows it all.

He must learn from the experiences, the achievements and failures, the observations and deductions of other men.

He must be teachable.

And since the opportunities for personal observation and for personal contact and converse with those who are capable of imparting information are limited, it is necessary that much of this shall be absorbed from their written word.

Notwithstanding the opprobrium that attaches to the "book engineer," every well-informed man owes much of his equipment to his reading.

Think it over and see how many of the things that have been of the most use and consequence to you, you learned out of your books and papers.

F. R. Low

Calculating Expansion and Aligning Steam Pipes to Large Turbines

PIPE alignment has become an increasingly important factor in the smooth operation of all large turbines. The instructions of the Westinghouse Electric & Manufacturing Co., which has for many years strongly advocated careful adjustment of steam connection to throttle valve, are here substantially presented.

DESIGNING the steam connection from a header to a large steam turbine requires proper engineering consideration, since stress due to piping and turbine expansion must be restricted to a definite value. As turbine capacities are increased, the piping becomes relatively stiffer. Large turbines are relatively less able to withstand external stresses than small sizes of fewer stages. Designing the steam connection for sufficiently low expansion stress and proper flexibility, as well as checking these conditions in erection, is therefore of prime importance for all types of large turbines.

The expansions of which cognizance must be taken are those due to the turbine itself, from cold to the maximum operating temperature, which is readily known. This extreme range of temperature need not be taken account of in the steam line, as it is not important if the turbine be stressed considerably when the steam is off the line and the turbine at rest. Calculations, therefore, are based on the expansion of the steam line due to the temperature difference between the minimum working pressure of saturated steam and the maximum temperature when superheated.

The flexibility of the steam pipe is to be obtained by having elements at right angles to each other. There should be such elements of sufficient length in close proximitiy to the turbine (one of them being vertical or not as is most convenient for the piping layout), and the respective lengths of this piping should be so adjusted that the piping will be deflected an approximately equal amount in either direction from the normal, as it is by doing this that the least stress will be imposed. Greater flexibility is secured by straight piping at right angles than by corresponding pipe made up of big radius bends.

When figuring pipe expansions, steel should be taken as expanding 0.000,0065 in. per deg. F. temperature difference per one inch length. The corresponding figure for cast iron should be taken as 0.000,0059 in.

It should be seen that each section of pipe before being put in place is thoroughly wiped out with suitable brush or swab. Such blowing of steam as can be permitted in a power house has little effect on large pipes.

In order to prevent stresses due to pipe expansion under full load conditions, the necessary adjustments are made with the turbine cold and piping heated to

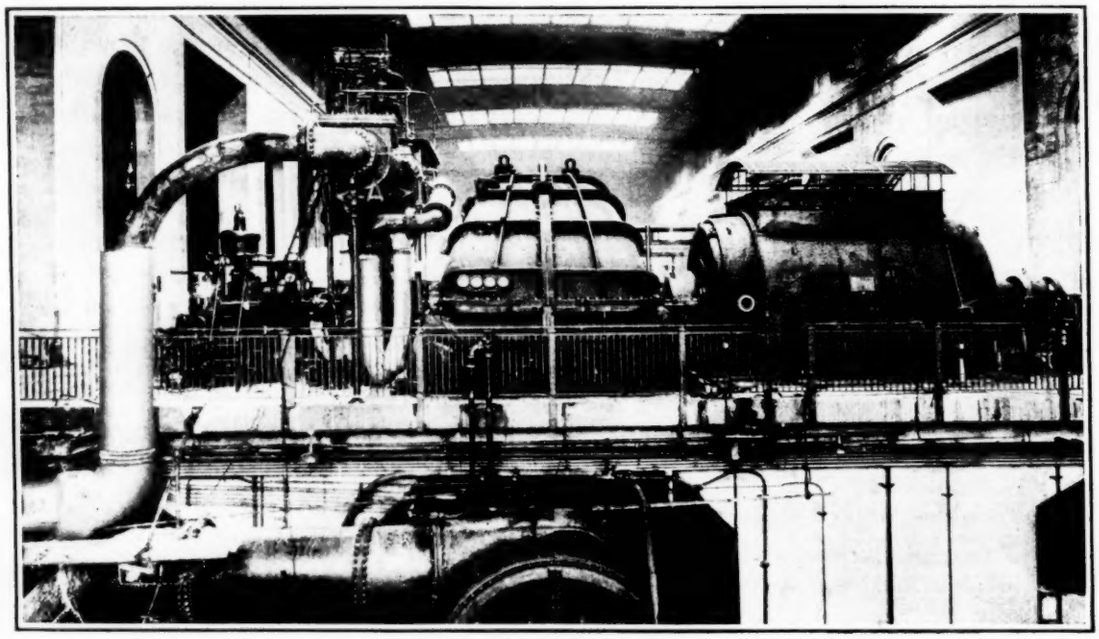


Fig. 1—Throttle valve with spring supports A on each side may be disconnected from steam chest for checking alignment

saturation temperature. If the closed throttle valve be disconnected from the steam chest and flexibly supported, as is possible in the turbine, Fig. 1, the line can be filled with saturated steam at normal pressure, and the position of the valve flange relative to the steam chest measured.

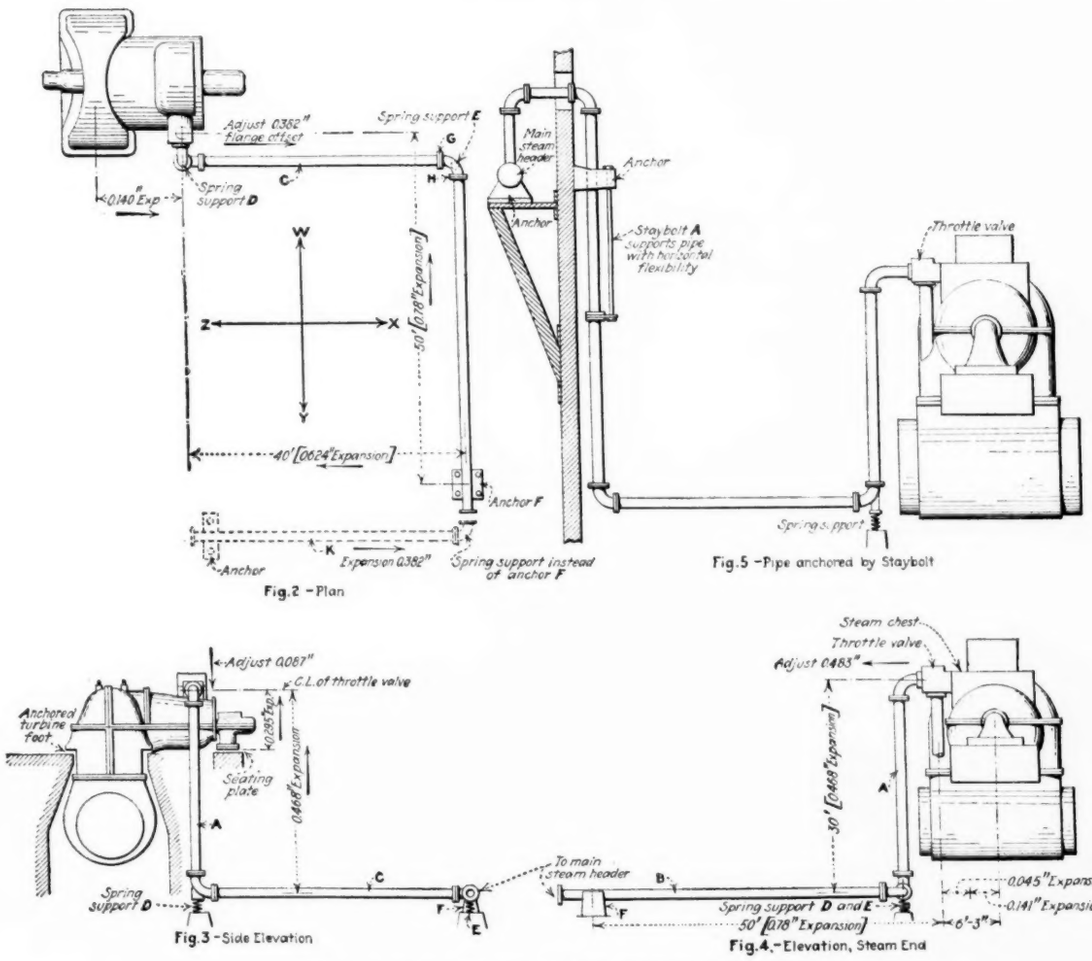
Calculations can be made to show what the expansive movements of both valve and turbine will be from these conditions to those of full load. The piping should be adjusted so that after such expansion takes place, the throttle valve flange will be in the desired alignment with the turbine steam-chest flange.

Suppose for illustration that from the condition of adjustment to the condition of full load (from saturated to superheated temperature) the steam pipe as at *A*, Fig. 4, should expand vertically upward $1\frac{1}{2}$ in. at the throttle-valve flange center line. Assume also that turbine expansion (from cold to full load temperature) should move the steam chest upward $\frac{1}{2}$ in. The pipe movement will therefore exceed that of the steam chest by $1\frac{1}{2} - \frac{1}{2} = 1$ in.

If the throttle valve were set 1 in. lower than the steam chest, under adjustment conditions, it would be central with the steam chest when the turbine is exactly at full load. That is, the one inch difference in expansions would be compensated for under adjustment conditions, so that the movements due to superheated temperature on the pipe and full load on the turbine would bring the throttle flange at the same elevation of the turbine flange, producing no strain vertically on the turbine when bolted together.

At an overload, however, the turbine would be somewhat warmer than at full load. The pipe which does not expand more would then tend to pull downward on the steam chest. If the turbine were run non-condensing, it would be still warmer, producing a somewhat greater strain due to pipe bending. In practice, therefore, it is customary to set the throttle valve so that at full load it will tend to be a little higher than the steam chest. Then there will be less pipe strain when running at overload or non-condensing.

If the throttle were set one-half inch below the steam chest for the adjustment condition, it would tend to be one-half inch above at full load and less than this at overload or non-condensing. Therefore, in all expansion adjustments, one-half of the calculated movement from conditions of adjustment to those of full load are used for the compensating when being adjusted. If steam chest and throttle valve tend to move in the same direction as just described, one-half the difference of these displacements is the correct adjustment. If, however, the valve and steam chest tend to move in opposite



Figs. 2 to 5—Three piping layouts illustrated

Figs. 2, 3 and 4 show expansions from conditions of adjustment to those of full load. Arrangement as at *K*, Fig. 2, simplifies expansion. Fig. 5 shows pipe anchor with horizontal flexibility.

directions, such as toward each other or away from each other, one-half the sum of the movements is taken as the required setting, as hereinafter described.

As an example of adjusting turbine pipe connections, consider the arrangement, Figs. 2, 3 and 4, which include three lengths of pipe. Section A, 30 ft., Section B, 50 ft., and Section C, 40 ft. long, are at right angles to each other, the pipe being carried on spring supports at points *D* and *E*, as shown. The supports at *D* and *E* should swing freely so that there is freedom for movement in a horizontal plane. The piping may be rigidly anchored to the main foundations at *F* or some other location farther away from the turbine, if added flexibility may be secured thereby.

Working Out Expansion Allowances

The expansion allowances are worked out for designed operating conditions of 300 lb. per sq.in. gage steam pressure, 200 deg. F. superheat, the range of temperature for the connections between the steam chest and the turbine will be from 80 deg. F. to 622 deg., a range of 542 deg.

The total vertical upward turbine expansion between the seating cover plate, Fig. 3, and the center line of the throttle valve from cold to full load temperature will be 0.295 in., which is the expansive movements of three elements. First, the vertical length of steel parts acted upon between the throttle-valve center line above the turbine and the primary nozzle chamber inlet below, is 35 in., the total vertical expansion of which is $35 \times 542 \times 0.000{,}0065 = 0.123$ in. Secondly, the distance between the primary-nozzle-chamber inlet flange and the turbine bearing center line is 52 in., the temperature range being from 80 to 551 deg. F., and the expansion is $52 \times (551 - 80) \times 0.000{,}0065 = 0.159$ in. Third, the vertical expansion of the cast-iron thrust pedestal based on 40 deg. rise and 55.75 in. height will be $55.75 \times 40 \times 0.000{,}0059 = 0.013$ in. The three turbine expansions will total $0.123 + 0.159 + 0.013 = 0.295$ in. upward, as in Fig. 3.

With regard to the expansion of pipe *A* from saturated to superheated steam, the range of temperature will be between 422 deg. and 622 deg. F., a range of 200 deg., and the expansion corresponding to this temperature difference is $200 \times 30 \times 12 \times 0.000{,}0065 = 0.468$ in. The pipe therefore expands upward more than the turbine, the latter being 0.173 in.

The difference between the vertical expansion of pipe *A* and the vertical expansion of the turbine equals the amount that must be taken up at full turbine-load by bending downward the pipe *B*, or $0.468 - 0.295 = 0.173$ in. The steam piping should therefore be lined up with dry saturated steam at full pressure on the steam pipe, so that the center of the throttle-valve body will be one-half this amount, or 0.087 in. below the center line of the steam chest, as shown in Fig. 3.

In other words, at the time of starting, with the turbine comparatively cold, the turbine must bend the piping, raising *B* 0.087 in. vertically upward, whereas, under full-load conditions, the steam piping will be depressed vertically downward the same amount, reducing the vertical force to a minimum. At lighter load a point may be reached where the pipe *B* is neither raised nor depressed, with no vertical stress on the turbine.

After allowing for vertical expansion as described, the horizontal expansion, 0.78 in. of the pipe *B* in the direction of the arrow *X* must be considered. This will tend to push the throttle valve into or against the turbine. There is also the turbine movement in the direction of arrow *Z* pushing against the pipe *B*, as given in Fig. 4, and figured from both the valve and turbine horizontal expansion. Since the pipe and turbine expand in opposite directions, one-half the sum of the expansions represents the distance for adjustment.

The distance 35 in. between the valve vertical center line and flange at a 200 deg. range gives 0.045 in. expansion. The distance 40 in. from the flange to the center of the steam chest increases 0.141 in., with a range of 542 deg. The sum of these three expansions, $0.78 + 0.045 + 0.141 = 0.966$ in., which distance the throttle-valve flange and steam-chest flange will tend to move toward each other. The adjusted distance is one-half, or 0.483 in., representing the amount which the two flanges should stand open, under condition of adjustment.

Calculating Throttle-Valve Adjustment

The throttle-valve adjustment for expansion of section *C* may be calculated in a similar manner. The turbine and pipe here expand toward each other as indicated in Fig. 2. Since the movements are in opposite direction, one-half the sum represents the adjustment distance by which the center lines of throttle valve and steam chest should be separated horizontally.

Summing up the foregoing paragraphs, for the arrangement given the steam piping should be supported as indicated and lined up as described, so that with the joint between the throttle valve and the steam chest broken, the flanges will be parallel, the throttle flange will be 0.48 in. away from the steam-chest flange, its center line will be 0.382 in. toward the steam end and 0.087 in. below the center of the steam-chest flange.

Note that the expansion of the pipe and the turbine are sometimes added and at other times subtracted. The former is done when the expansion of the turbine is opposite in direction to that of the pipe; the latter, when the expansion of both are in the same direction and partly cancel each other. In this case the one expansion is subtracted from the other. In both cases the offsetting of the pipe is one-half of the amount so that, as said before, the pipe will be bent an equal amount in opposite directions.

When bolting up the throttle valve, it will of course be necessary to spring it in place, which may be done with confidence that the best arrangement has been secured.

In the case of the 20,000-kw. turbine, Fig. 1, the steam chest is bolted rigidly to the turbine cylinder, the throttle valve partly supported by springs and partly by being bolted to the steam chest.

Final Adjustment of Steam Pipe

For the final adjustment of the steam pipe the throttle valve will be bolted up to the steam pipe with joint made up and with the joint broken between the throttle valve body and the steam chest. Square-head studs are provided to enable them to be removed for paralleling the joint and inserting the gasket.

Before finally bolting the throttle to the steam chest, check up to be sure the pipe has ample flexibility up and down and in every direction in a horizontal plane. The flange should stand parallel but be sure the flange

may be freely sprung some amount, say roughly, one degree, out of parallel in every direction.

One may be misled by the apparent flexibility of a pipe by being able to push the free end of a disconnected pipe, when the flange does not move parallel. In practice, the flange must move parallel and therefore the pipe must bend as *B* and assume a double curve with consequent loss of flexibility, requiring eight times the force to produce a given deflection.

The steam piping should not impose a greater load than an amount specified by the builder on the throttle valve flange in any direction in any of the extremes of the range of operating conditions. The flexibility of the steam piping must be studied with this limit in view.

When considering the deflection of the steam piping in order to determine the loads imposed on the throttle valve for any given layout, note that, for the conditions given, if the turbine goes non-condensing, the expansion between the coupling end foot and the center of the throttle valve will increase from 0.140 in. to 0.181 in.

Adjusting Steam-Chest Supports

After the joint between the throttle valve and the steam-chest body has been bolted up, the steam-chest support springs should be adjusted to give a total compression of $1\frac{15}{32}$ in. with the turbine cold. The weight of the throttle valve is 10,750 lb. and the combined scale of the steam-chest support spring is 8,000 lb. per in., in the assumed case, so that the spring compression required to support the weight of the throttle valve is $1\frac{11}{32}$ in. The calculated difference in expansion between the center line of the throttle valve body and the top of the steam-chest support spring struts is $\frac{1}{4}$ in., so, in order that the force imposed on the steam-chest inlet flange at the extreme limits of expansion is a minimum, the springs are given an additional $\frac{1}{8}$ in. compression with the machine cold, making a total compression of $1\frac{15}{32}$ in.

An alternative piping arrangement is shown as dotted, at *K*, Fig. 2, which has much to commend it inasmuch as the expansion of *C* and an equal portion of *K* cancel each other. The anchor may be offset some amount from the turbine throttle so that the expansion in the greater length of pipe *K* will compensate for the expansion of the turbine.

Another alternative is shown, Fig. 5, where the line should be supported by strong tierods as at *A*, which support the steam-chest line at an elevation corresponding to about the center line of the turbine, and in such position that the vertical expansion of one member cancels the vertical expansion of the turbine and the other member.

The tierods must permit free motion in a horizontal plane. The portion of pipe below the tierods is supported on springs where necessary to take the weight. There should be an anchor on the main header adjacent to the point where the turbine line connects, to preclude any axial motion of the header at this point.

It is not to be expected that steam lines may be taken down and bent or otherwise changed to give the fine adjustment required. It is a convenient process in the first instance to insert steel dutchmen, as, for example, at *G* and *H*, Fig. 2, these being arranged so they may be thinned after the observation of the setting of the throttle valve flange so as to bring the flange without further change to the definite required position.

It sometimes happens, as for instance, if the steam line came from the opposite side of the machine from that shown in these sketches, that the calculations show that the adjustment of the throttle and steam-chest flange instead of being some amount apart, should be within each other by some amount. Obviously, in this case the observation must be made with the flanges separated some amount, so a dutchman of convenient thickness is inserted in the pipe, which, after observations are complete, is thinned a definite amount.

It is the custom to invite purchasers to submit their piping plans for comment and criticism. These are freely given to the end that the installation may be the more satisfactory. When it comes to the actual installation, it is frequently possible for the erecting engineer to discover means of improving the arrangement to secure a greater degree of flexibility.

Correct Steam-Cylinder Lubrication

A brief summary of those factors to be considered if correct steam cylinder lubrication is to be obtained are:

A lubricating film is most easily formed when the cylinder oil is comparatively atomized.

Complete atomization depends upon steam temperature and moisture content, upon steam velocities and mechanical atomizing appliances, upon viscosity and surface tension, and upon the time allowed before the oil reaches the cylinder.

High temperatures improve the atomization of the oil; low temperatures retard it and make it necessary to use quick-atomizing oils.

Mechanical atomizers and a suitable arrangement of the feed lines can be made to improve the atomization of an oil. They are sometimes quite necessary for the proper atomization of viscous oils.

Low-viscosity oils are more readily atomized than high-viscosity oils.

Oils of low surface tension are more completely atomized than those of high surface tension.

Filtered steam-cylinder oils atomize more quickly than those that have not been filtered.

Moisture in the steam improves the atomization of the oil.

Compounding in the oil improves its ability to atomize.

High steam velocities improve the atomization of the oil, although they allow a shorter time for complete atomization before the oil reaches the cylinder.

The calorific value of a combustible substance is the number of heat units evolved when a unit weight of the substance is completely burned. It is an expression of the number of calories evolved by a kilogram or the number of B.t.u. evolved by 1 lb., the first figure being convertible to the second by multiplying by 1.8 the ratio of the length of the Centigrade to that of the Fahrenheit degree.

Although a certain amount of air admitted to the furnace above the fire is usually essential for good combustion, air leaks due to faulty furnace fittings and cracks in the boiler brickwork are often responsible for the poor draft and consequent inefficiency of the plant.

Effect of Clearance on Air-Compressor Operation

CHARTS enable one to find the volumetric efficiency and mean effective pressure of a compressor without computation. By adjusting clearance, load may be changed at will.

THE effect of clearance on the output of an air compressor and upon the power required is not appreciated by many. Theoretically, clearance should not influence the net efficiency of the machine but actually it does, for a larger cylinder is needed for

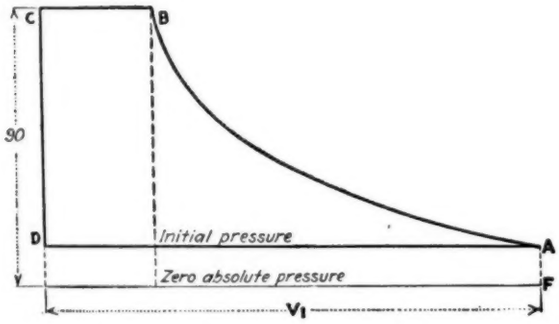


Fig. 1—Ideal no-clearance diagram

a given capacity over that required in a non-clearance machine. This will naturally result in a larger friction loss and will call for a greater power input.

An ideal air-compressor diagram is shown in Fig. 1. Here air is drawn into the cylinder during the suction

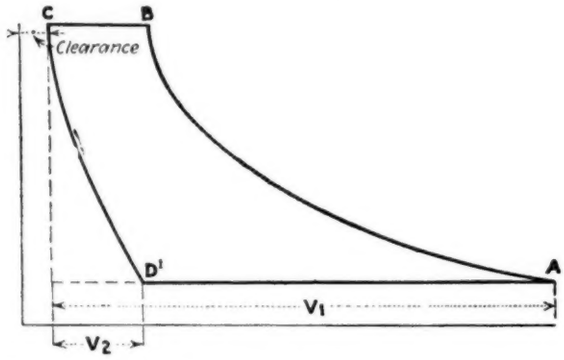


Fig. 2—Effect of clearance on an air compressor

stroke DA. Compression of the air takes place along AB, and the air is discharged along BC. The reversal of the piston's stroke causes the pressure within the cylinder to drop instantaneously to D, the suction pressure. The volume of air drawn in is represented by the length V_1, and since this volume is equal to the piston displacement DA, the volumetric efficiency is unity. The mean effective pressure, or the average pressure exerted on each square inch of the piston face during one complete cycle, is proportional to the area of the diagram divided by its length.

In every compressor a small amount of air remains in the cylinder when the piston reaches the end of its strokes, filling the clearance cavities about the valves and the space between the piston and cylinder head. This air expands behind the piston as the latter starts on its suction stroke, and the line CD in Fig. 1 curves to the right as shown by CD^1 in Fig. 2; by the time the clearance air pressure drops to a value equal to the suction pressure, it fills that part of the cylinder represented by V_2. It is plain that only $V_1 - V_2$ volumes of fresh air enters through the suction valve. The volumetric efficiency is then $\frac{V_1 - V_2}{V_1}$

The diagram area is less, due to the re-expansion, and consequently the mean effective pressure is less, and the amount of air drawn in is decreased by the existing clearance. A study of the diagram will reveal that with

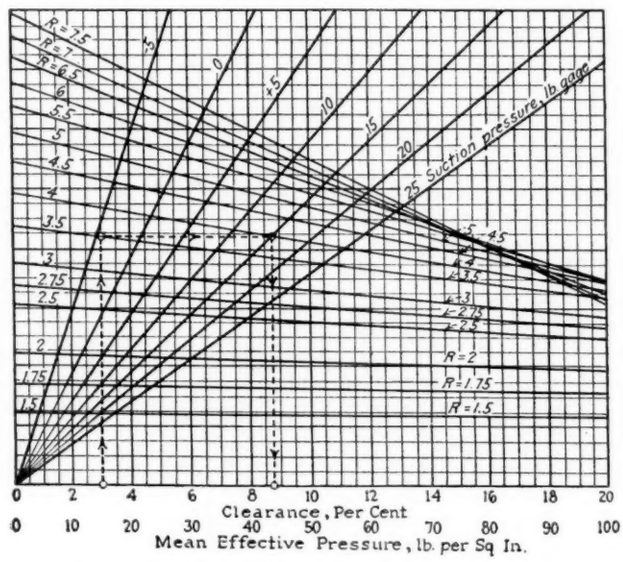


Fig. 3—Chart gives mean effective pressures

a high discharge pressure the re-expansion volume V_2 will be greater than in case of a low discharge pressure for any given clearance.

Disregarding heating of the air at the suction valves, jacket cooling and other variables, the volumetric efficiency depends upon the clearance and upon the suction and the discharge pressures. In Fig. 4 is given a chart by which the volumetric efficiency of a compressor operating under any suction or discharge pressure and with various clearances can be found. For example, assume that the compressor is connected to the discharge of a turbo-blower and takes in air at 15 lb. gage pressure,

discharging it at 90 lb. gage and has a clearance of 3 per cent. Starting at the 15-lb. suction pressure value in Fig. 4, move vertically until a horizontal line drawn from the 90-lb. discharge pressure value is cut; as shown, this point lies on the diagonal marked 3.5, which means that the ratio of the absolute discharge pressure to the absolute suction pressure, or $\frac{90 + 15}{15 + 15} = 3.5$.

This diagonal is followed downward to the left until the 3.5 line sloping toward the right is met. Moving along the latter line until a line drawn vertically from the clearance value of 3 per cent is met, a line is drawn horizontally to the left-hand scale, giving a reading of 96 per cent for the volumetric efficiency.

To determine the mean effective pressure, Fig. 3 is taken up and a line drawn vertically from the 3 per cent clearance scale until the proper pressure ratio line R is reached; in this case, as found by Fig. 4, $R = 3.5$. The pencil is then moved horizontally until the diagonal reading 15 suction pressure is reached. Moving downward from this gives a mean effective pressure reading of 43 lb. per sq.in.

As as example of the use to which the charts may be put, suppose one has a 6,000-cu.ft. compressor operating between a suction pressure of zero gage, or 15 lb. absolute, and a discharge of 90 lb. gage, or 105 lb. absolute. It is desirable that the plant output be increased from 6,000 cu.ft. of free air per min. to 10,000 cu.ft. Two solutions are possible. A second reciprocating compressor of 4,000 cu.ft. may be installed or a centrifugal, or turbo, blower may be connected to the suction of the present compressor. The blower should have a free-air capacity of 10,000 cu.ft., delivering 6,000 cu.ft. of air at 15 lb. gage to the reciprocating compressor, which would then compress the air from a pressure of 15 lb. gage to 90 lb. The sole point of debate as to the advisability of installing the blower is whether the steam cylinders of the present compressor could operate under the new air-cylinder conditions without such a long cutoff as to make the engine inefficient. By Fig. 4 we find that R, the ratio of absolute pressure of 0 and 90 lb. gage, is 7. With 3 per cent clearance, by Fig. 3, the present mean effective pressure in the air cylinder is 35 lb. per sq.in. Increasing the suction pressure to 15 lb. gage, or 30 lb. absolute, changes R to 3.5. With the same clearance the m.e.p. increases to approximately 43.5 lb. This is an increase of almost 25 per cent, making the cutoff of the steam cylinder much later than at present and increasing the water rate to a marked degree.

In all probability an increase of 10 per cent in the load would not increase the steam consumption per horsepower to any extent. In other words, the air cylinder m.e.p. can be raised from 35 to 38 lb. Working backward from 38 lb., it is found by Fig. 3 that the clearance must be increased to 12 per cent. But by Fig. 3, with 12 per cent clearance and $R = 3.5$, the volumetric efficiency is 82.5 per cent, while under present conditions with 3 per cent clearance and 0 gage suction pressure, the efficiency is 92.5 per cent. The volume of air actually handled by the compressor would be approximately 10 per cent less than with a 3 per cent clearance, and instead of handling 6,000 cu.ft. it will handle only 5,400 cu.ft. The turbo-blower, when taking in 10,000 cu.ft., will deliver about 6,000 cu.ft. of air at 15 lb. gage. By cooling this air back to the atmospheric temperature, the product of the absolute pressure and volume will be equal to the product of the volume and absolute pressure of the air taken into the blower, or $P_1 V_1 = P_2 V_2$. In the present case we can write $(15 + 0)\ (10{,}000) = (15 + 15)\ (5{,}000)$, or the volume taken in by the reciprocating compressor need not be much over 5,000 cu.ft. Consequently, the addition of extra clearance volume to hold the m.e.p. to 38 lb. will not lower the ultimate plant capacity over that desired. The extra clearance may in some compressors be obtained by placing spacers between the cylinder and heads, and in some machines shorter pistons can be used.

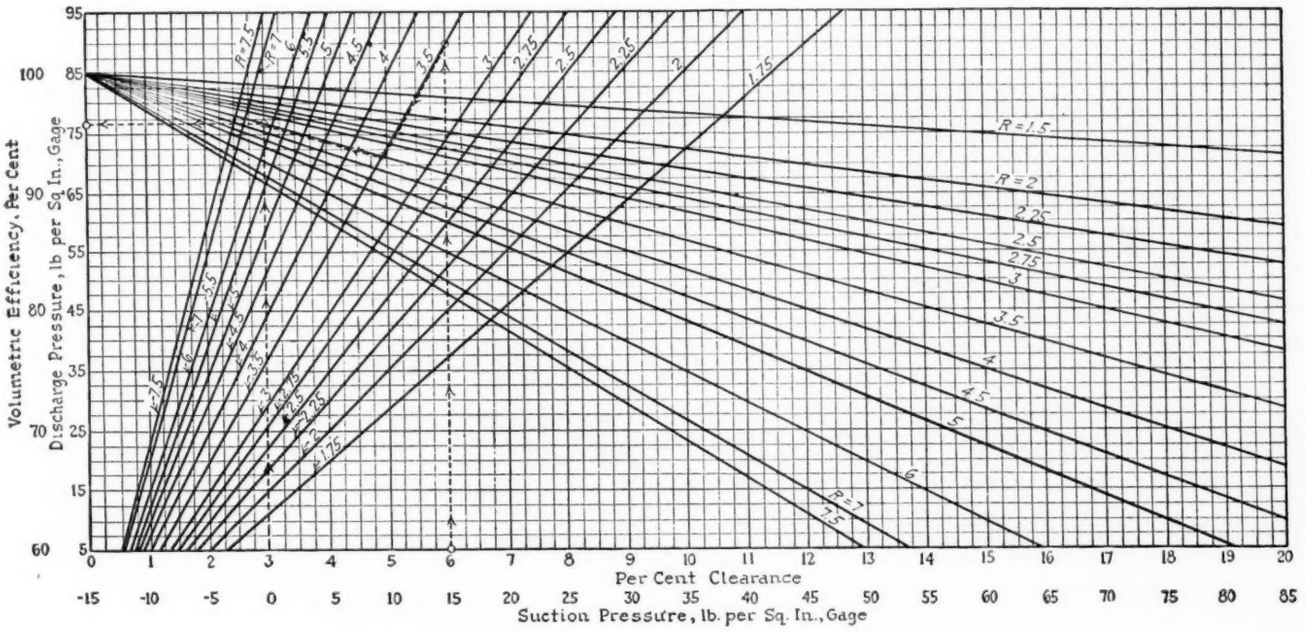
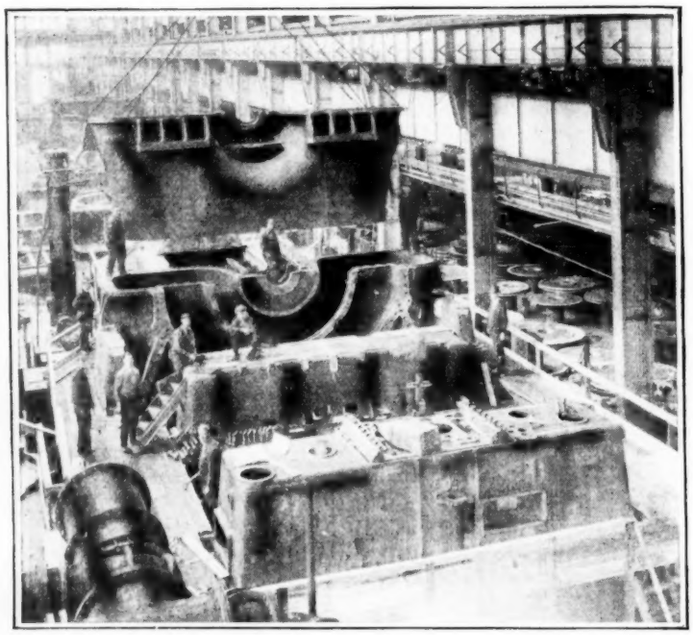


Fig. 4—Compression ratios and volumetric efficiencies

Assembling Cylinder of 50,000-Kilowatt Turbine



One of two 50,000-kw. turbo-generator units being built for the new Richmond station of the Philadelphia Electric Company. These are of the tandem-compound type with double-flow in the low-pressure cylinder. They are designed for 375 lb. steam pressure, 230 deg. superheat and one inch absolute back pressure. The cuts show successive steps in the shop assembly of the double-flow low-pressure cylinder and high-pressure exhaust hood.

Photos by courtesy of General Electric Company.

Fig. 1 (Left)—Lower half, turbine end, of the low-pressure exhaust hood is being dropped into place beside the lower half, generator end, of the same hood

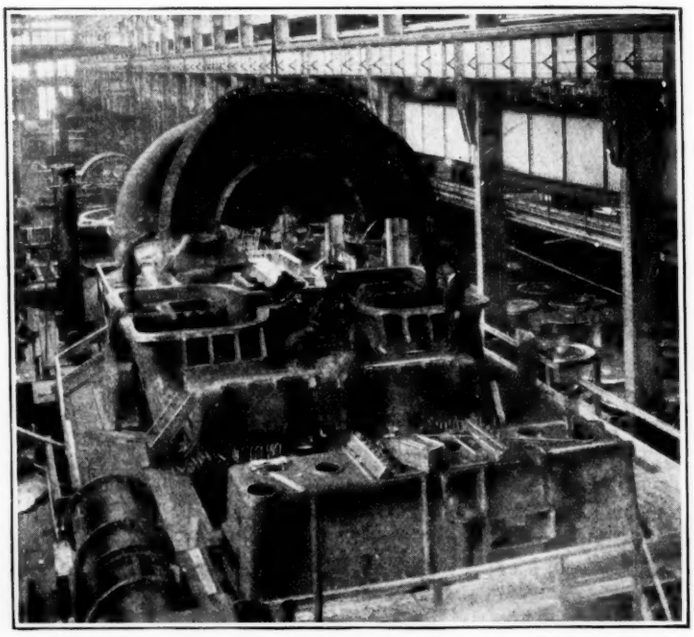


Fig. 2 (Right)—Lower half of the low-pressure exhaust hood is in place and the generator end of the upper half is being dropped into place

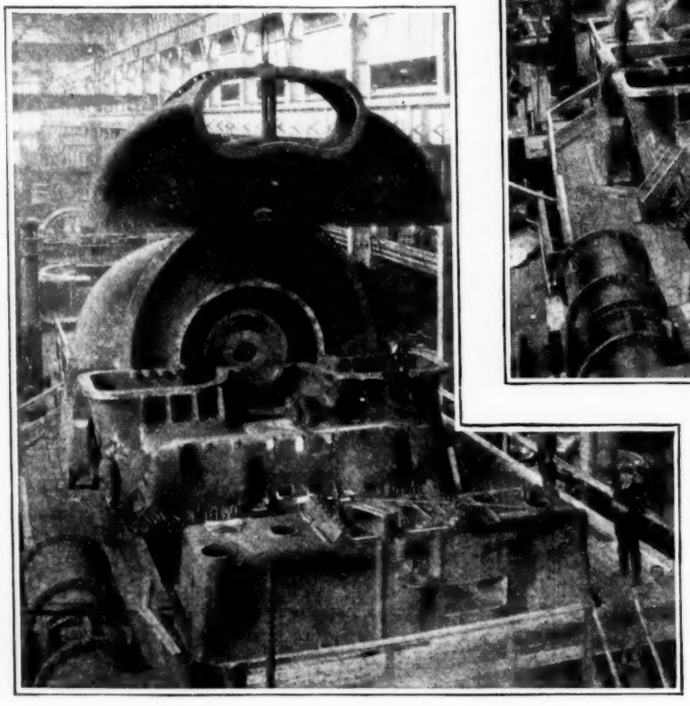


Fig. 3 (Left)—Dropping the upper half, turbine end, of the low-pressure exhaust hood into place

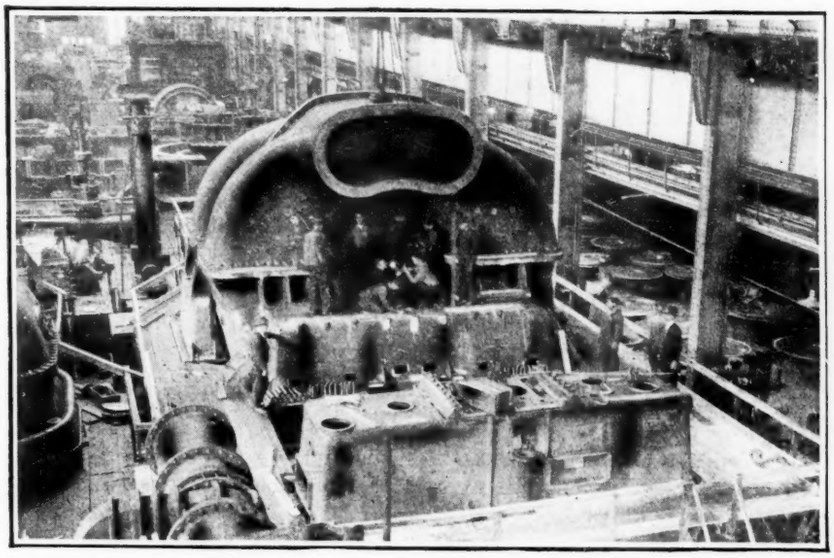


Fig. 4 (Right) — The low-pressure exhaust hood assembled. Tests are being made for alignment, etc.

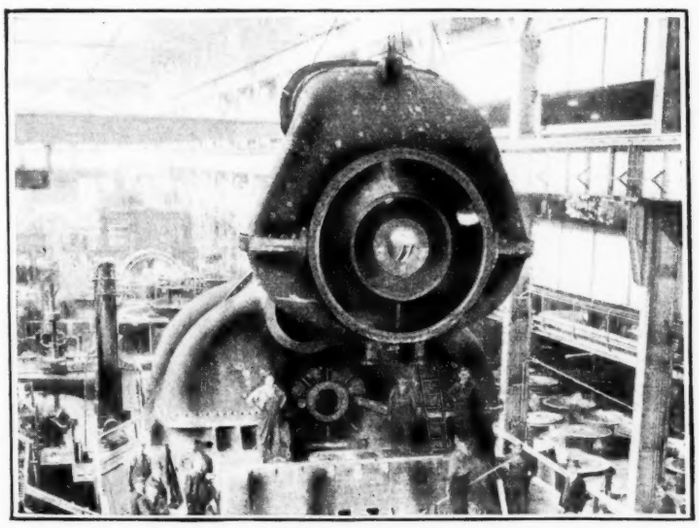


Fig. 5 (Left)—Here the crane is lowering the high-pressure exhaust hood into place. Note the large area of the connection through which the steam exhausts to the low-pressure hood

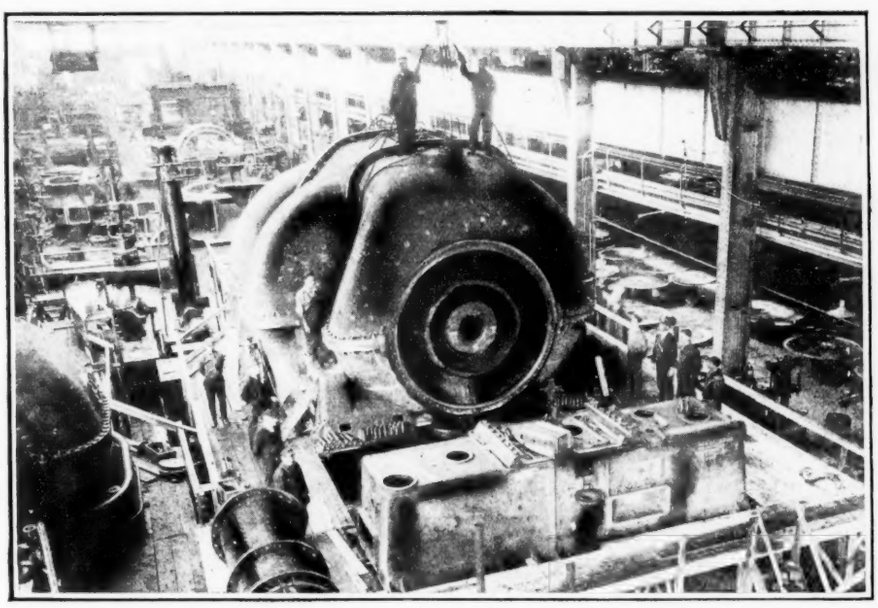


Fig. 6 (Right) — High- and low-pressure exhaust hoods assembled in place

Saving Coal by Correct Hand Firing

AN ENGINEER relates some personal experience in hand-fired plants. Finds that firemen do not understand combustion. Change in firing methods reduces coal bill.

By J. A. MADSEN

ALTHOUGH most modern power plants are equipped with some form of mechanical firing of boilers, there are thousands of older plants using over ten tons of coal per day where hand firing is still followed. In many of these plants the firemen have received no instruction in the correct method to follow with a given coal, nor are they given any inkling as to the steps involved in obtaining proper combustion. With a little attention given to such boiler-room forces, the reduction in coal consumption would be more than startling.

To illustrate this point that the problem of training firemen is not being given proper attention, I am prompted to relate experiences in two hand-fired plants.

I once had charge of a plant equipped with four Manning boilers, running three 8-hour shifts with one fireman per shift. Three boilers were sufficient to carry the load, one being down for cleaning and inspection each week. The boilers were operated on natural draft, and the average steam flow was 15,500 lb. per hour with an evaporation of about 7.8 lb. of water per pound of coal. Inasmuch as a high-grade coal was being used, I felt that the performance of the boiler plant could be bettered and it seemed that the method of firing was at fault. Of course the combustion space in this type of boiler is too small to give the best results, and to secure good combustion these boilers call for firing methods somewhat different from those with a return-tubular or a water-tube boiler.

No Firing System Followed

The firemen followed no definite method of handling the fires, simply leaving them alone until the steam pressure dropped about five pounds, and then breaking up the fuel bed with the slice bar or rake. Fresh coal would be added regardless of whether the fire was bright or not. Even though these boilers were not overloaded, the firemen seemed unable to hold the main damper closed for any length of time, although it closed and opened on a pressure variation of two pounds.

One Monday morning I decided to handle the fires for an hour to see if there was any serious obstacle in the way of possible improvement, before criticizing the firemen concerning their firing methods. The fires were about 18 in. deep at the front and in the center, but after leveling with a rake I found that holes existed all about the edges, admitting large quantities of excess air at these points and reducing the available draft. After all the fires were leveled to a uniform depth, they were allowed to burn down to a thickness of about 8 in. and then each furnace was charged with three shovelfuls of coal. The entire grate area was not covered with a blanket of green coal, as this would reduce the furnace temperature below the ignition point of the gases and cause a temporary deficiency in the air supply, just when the greatest amount of air should be admitted. But by placing each scoop of coal so it was surrounded by a hot zone of incandescent fuel, the temperature was maintained at a higher point and better combustion resulted. The amount of air entering the furnace was greatly reduced as the uniform depth of the fuel bed cut down the air entering at the sides and allowed the center of the fuel bed to get its proper amount of air.

The damper closed more quickly with this method of firing and we endeavored to hold the damper in the closed position as long as possible, not waiting for the steam pressure to drop before attending to the fires. The firemen had formerly complained that the draft was not sufficient for the load, but after some experimenting they found that with the lighter firing and uniform depth of fuel bed, it was actually possible to close the hand damper slightly on the boiler nearest the stack, thus equalizing the draft on all boilers.

Savings Made

The evaporation was increased from 7.8 to 8.75 lb. of water per pound of coal, making a reduction of 2½ tons per day in our coal consumption. The grade of coal that we were using cost $6.80 per ton in the bin, so a yearly saving of $5,610 was effected by the simple expedient of finding the correct method of firing for that particular plant and the expenditure of a few hours' time instructing the firemen.

In another of this firm's plants, equipped with six return-tubular boilers, the firing job was considered to be very hard and undesirable, a fireman staying an average of but two weeks. The chief engineer had a few months previously changed to a cheaper grade of coal, but the firemen had great difficulty in holding the steam pressure. Inasmuch as a large sum had been spent for new dumping grates, the firm would not consider going back to the more expensive fuel.

The former fuel, a low volatile coal, had been high in fixed carbon and low in ash, while the coal being used ran high in ash and sulphur, with a high percentage of volatile. A stoker would have been the logical thing, but we were confronted with the task of burning this coal on hand-fired grates and maintaining a steady steam pressure. The firemen made no attempt to experiment and find a way out of their difficulties, but fired in the same manner as they had done with the other coal.

Suggested Firing Method

I at once suggested the alternate plan of firing, using only three or four scoops of coal at one firing. The fireman on the day shift shook his head, saying that the system had been tried and it was found to keep a man busy at the firedoors all the time.

As I stood watching them, both firemen started to cover the fires of the boilers at the same time. Each man cared for six firedoors and put eight scoops of coal

through each door. The scoops held at least 20 lb. of coal, making 1,920 lb., or nearly a ton of coal at one charge. After taking a look at the water in the gage glass, they went to the door to cool off. The steam pressure dropped as soon as the fires were covered and in about five minutes the firemen had to pick up the slice bars and start slicing the fires, which were about 15 to 18 in. thick. The pressure then started to climb and they went back to the door.

After a while the pressure dropped again and I opened a firedoor to see the condition of the fuel bed. The fires were too heavy to allow a clear view all the way back to the bridge wall, but the condition of the fire at the front was such that a radical change in firing methods was imperative. At the points where the bar had been raised up through the fire it was white hot, but all other parts of the fire were dull, indicating a lack of air, and holes were beginning to show up at the bright spots. I ordered the firemen to level the fires and push the fire back from the door with the rake. After this operation the fires grew bright and the firemen prepared to fire up again.

"Now, boys, you have had to work hard when doing it your own way, and I want you to try my way for awhile," I informed them. "According to my way of doing the firing, only one man will have to be in front of the boilers at a time, the other man can cool off and rest at least half the time. I will guarantee that my way will prove easier than the way you have been working."

Fireman Had More Leisure

I told one man he could sit at the door while I started the other on the new system of firing. I had him fire four small scoops of coal into one door of No. 1 boiler and repeat this process on one side of each of the other furnaces. Then he returned to the first boiler and covered the remaining half of the fire, which was now at a white heat. The high furnace temperature maintained by keeping half the fire bright, prevented the usual drop in the pressure when covering fires. I had timed the operation of covering the fires and found that it did not exceed five minutes. By the time the fireman had reached the door, the steam pressure was high enough to close the damper and both of the men looked pleased and were willing to admit that the system had merits.

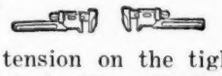
In about five minutes more the damper regulator lever began to oscillate, and I sent the other fireman to level all fires before the damper had opened wide, as the effect of a hole in the fire is more serious with full draft on the furnace than when the draft is checked. As he finished leveling the last fire, the damper again closed. This was a surprise to him. Adjusting the feed valves, he resumed his seat at the door after less than five minutes of work on the fires. Then before the fires got too thin, the first firemen was instructed to cover them in the way I had shown him. Both men admitted the work was easier, since no slicing of fires was required and the light fire did not cake over, and the draft appeared to be much better. I could see that they could not understand the real reason for all this. So I enlightened them as follows in simple non-technical language.

"You men had a hard time burning this coal because you overlooked a few important points. First, for every ton of coal that you shovel into the furnace you must supply 17 or 18 tons of air, and the most air is needed at the time the coal is fired. But when you fired such heavy charges of coal, you reduced the air supply just when you needed it most and cooled the fire so low that very little steam was generated. In order to get any air at all, you had to do a lot of hard work with the slice bar. After you had leveled your fires, you had a good bed of coke which made a bright fire and then you got plenty of air.

"With the new system, when you fired a light charge of coal on one side," I continued, "the air supply was not choked, as you had plenty of air coming through the bright side, and the volatile or gassy part of the coal was burned because you had a high temperature and a good mixture, two things you must have to burn this grade of coal. Your work is easier than before, since one man is resting while his mate is working, which proves that each of you works only half of the time, and besides you have much less coal to handle."

Bonus System Adopted

They readily saw the logic of this statement and viewed the task in a new light. From then on no trouble was experienced in keeping the personnel of the boiler room intact, and the coal consumption dropped from 36 tons per day to about 30 tons. This was a clear saving of 16 per cent in the fuel bill, and the management was favorably impressed with what had been accomplished. At the urging of the chief and myself, they paid a bonus to the boiler-room crew at the end of the month. This proved to be an incentive and the coal consumption was further reduced, as we found that the plant could be operated on four boilers at night when the load was less than on the day shift. In the beginning the firemen knew nothing about the ratio of air supply to the coal supply and the varying characteristics of coal, but once they were given a little education along these lines, they used their knowledge to good advantage.



The maximum tension on the tight side of a belt equals that on the slack side multiplied by 2.718^{fB} where f is the coefficient of friction between the belt and pulley and B the angle of contact measured in radians. One radian is equal to 57.4 deg. This formula is not particularly convenient for quick computations, but from it can be derived a simple rule easily memorized and handy for making quick comparisons in the field without the aid of handbooks or slide rule. The rule is this: $40 \div f =$ angle of contact in which the tension doubles. An example will make the application clear. Take a leather belt on a cast-iron pulley with a coefficient of friction $f = 0.35$. The tension will double in $40 \div 0.35 = 115$ deg. (approximately). If the angle of contact is only 115 deg., the higher tension will be double the lower at the slipping point, leaving a net pull equal to the lower tension or to one-half the higher tension. Now suppose a weighted idler is applied increasing the arc of contact another 115 deg., making a total of 230 deg. This means doubling twice so that the high tension is four times the low. The net tension will be three times the low or three-fourths of the high. Assuming that the same high tension is used in both cases, the addition of the idler pulley increases the net pull from one-half to three-fourths of the high-tension, a gain of 50 per cent. This permits 50 per cent more power to be transmitted.

Locating Faults in Electric Elevators—Alternating-Current Motors

Types of Alternating-Current Motors Used in Elevator Applications—Effect of Voltage and Frequency Changes on Motor Operation—Causes of Various Troubles That May Occur in the Operation of the Motors

By Charles A. Armstrong

FAULTS in alternating-current motors used in elevator service will be the same as when these motors are used for other applications, except as the motor's operation may be affected by the conditions of the elevator machinery and the controller. With direct-current, shunt-type or compound-type motors are used for elevator service. In alternating-current elevator practice a wider diversity of motors are used and this somewhat complicates the problem of locating troubles when they do occur. Although single-phase type motors have been used to drive elevator machines, these have been exceptions to general practice, therefore only the polyphase types will be considered.

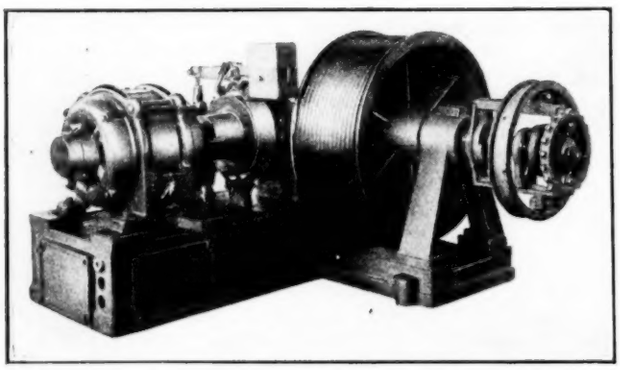


Fig. 1—Drum-type machine driven by single-speed squirrel-cage motor

Types of Alternating-Current Motors

On alternating-current circuits polyphase motors of both the squirrel-cage type, Fig. 1, and the wound-rotor type, Fig. 2, will be found in elevator service. For slow-speed machines up to about 200 ft. per minute the simple squirrel-cage or wound-rotor type motors are used. For medium-speed drum type or geared-traction machines, Figs. 3 to 5, multi-speed motors are used. These motors may have two windings in the same slots, each winding grouped for a different number of poles; for example, 8 and 4 poles or 12 and 4 poles, giving a speed ratio of 2 to 1 or 3 to 1 as the case may be. In some cases, only one winding is used, and this is regrouped for two different numbers of poles by the control equipment.

It is also the practice to use two different motors. The stators of the two motors may be mounted in the same frame and the rotors keyed to the same shaft so that the two motors form a single unit, Fig. 5. Control equipment is provided to cut into service first the slow-speed motor and after acceleration on this motor has been accomplished, the high-speed winding is switched into service and the slow-speed winding disconnected.

Where a two-speed wound-rotor motor is used, the rotors must have two windings grouped for the same number of poles as in the stator and these windings handled in the same way as the stator windings. On account of being unable to obtain proper speed control and the difficulties of designing and building very slow-speed alternating-current motors, direct-traction alternating-current elevator equipments are not used. Where it is desirable to go to high-speed elevators above 400 ft. per min. and only alternating current is available, direct-current equipment is generally installed and the alternating current is converted into direct current for the power supply.

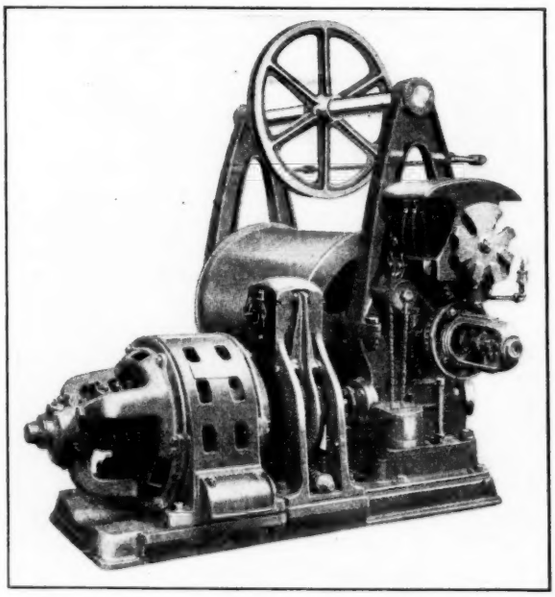


Fig. 2—Drum-type machine driven by wound-rotor motor

All the different types of alternating-current motors used for elevator service, referred to in the foregoing, affect the condition under which trouble may be experienced with these motors. Although the conditions under which the trouble may occur may vary somewhat with the different types of motors, practically the same difficulties may develop in all types. These troubles may be divided into: Noisy operation; motor fails to start; torque too low; speed too low; speed too high;

heating; sparking at the brushes on the slip rings, where the motor is of the wound-rotor type; and reverse rotation.

With alternating current, the quality of the power supply is affected not only by the voltage, but also by the frequency of the system. The torque of an alternating-current motor varies as the square of the voltage, therefore a comparatively small variation in

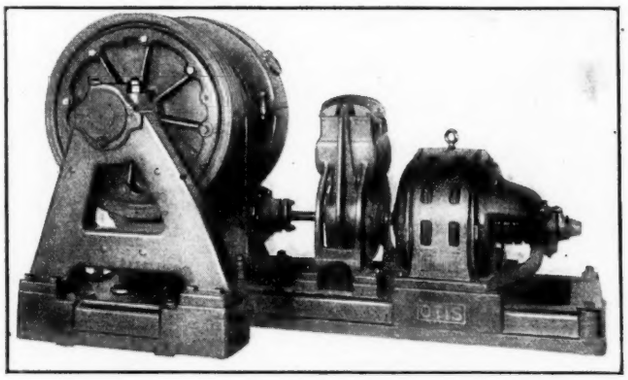


Fig. 3—Geared-traction machine driven by wound-rotor motor

voltage may materially affect the torque of the motor. For example, if the voltage is reduced 10 per cent on a motor, the torque it will develop will be only 81 per cent of that at normal voltage. It is quite readily seen that if the voltage is materially reduced, there is a liability of the motor's not being able to start its full load, or if it does start the load, acceleration will be slow. Where the elevator is making floor-to-floor stops, the greater part of the time is required to accelerate, therefore anything that lengthens the acceleration period will slow up the service.

Overvoltage on an alternating-current elevator motor may also have undesirable effects. A 10 per cent increase in voltage above normal will cause the motor to develop a torque 121 per cent of normal, which may increase the rate of acceleration of the elevator to a point where it will be unpleasant for the passengers in the car. It may also cause excessive noise in both the motor and controller. High voltage increases the current through the magnet coils and increases the pull on the contactors, which may cause them to slam unduly, resulting in increase in maintenance.

Checking Power-Circuit Voltage

When checking voltage on an elevator, it is not sufficient to take this reading at any particular load or time. What should be obtained, is a record for a period of a day or a week on a recording voltmeter. It is better still, if one of these meters can be connected to each phase for the test period. It is quite as important that the voltage be balanced on all phases as it is that it be correct on any particular phase. Where power is supplied over long transmission lines, unless automatic regulation is provided there may be wide variation in the voltage over the 24 hours of the day. In the daytime, when the load is heavy on the system, the voltage may be low and at night or during other light-load periods the voltage may be high. A variation in the voltage, applied to an alternating-current motor, from 10 per cent above normal to 10 per cent below normal, will cause the torque to be 50 per cent higher at the high voltage than at the low voltage; that is, 121 per cent torque is 50 per cent greater than 81 per cent.

With this wide variation in torque it may be impossible to adjust the control equipment to give satisfactory operation at all periods of the day. Adjustments that would give smooth acceleration at the low voltage may cause the motor to start with a jerk at the high voltage. On the other hand, adjusting the control for satisfactory operation at the high voltage may result in slow acceleration at low voltage, or the motor may actually stall.

Effects of Frequency Variations

If trouble is being experienced with the torque and speed of an alternating-current motor, the frequency of the power system should also be checked. The speed of the motor varies directly as the frequency. A 5 per cent increase in frequency will cause a 5 per cent increase in speed, and likewise a 5 per cent decrease in frequency will cause a 5 per cent decrease in speed.

A variation between 57 and 63 cycles on a 60-cycle system will cause a variation of about 10 per cent in speed of the motors on the system. An increase in frequency will tend to decrease the torque of the motor where a decrease will tend to increase the torque.

A change in frequency also affects the magnets on the controller. An increase in frequency will decrease the current through the coils and vice versa for a decrease in frequency. With a decrease in frequency

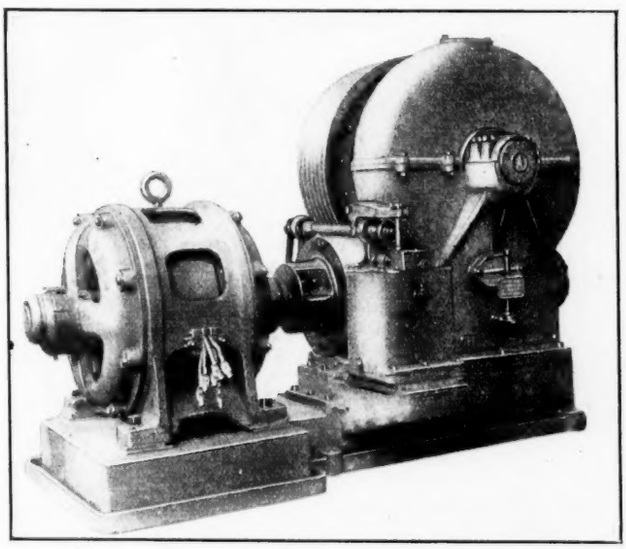


Fig. 4—Geared-traction machine driven by two-speed squirrel-cage motor

the increase in current in the magnet coils may cause excessive heating. From the foregoing it is evident that close voltage and frequency regulation are desirable for alternating-current elevator motors and controllers.

Noisy operation of alternating-current motors is more prevalent than with direct-current motors. As an alternating-current motor is inherently more noisy than the direct-current type, anything that might occur in the former to cause unusual noise may be doubly objectionable. In any motor the rotating member out of balance will cause vibration and noise in operation. With an alternating-current motor, in addition to

mechanical vibration there is also magnetic vibration to contend with, and the latter is more likely to cause trouble than that occurring from mechanical causes. Mechanical vibration, when once eliminated, is not likely to occur again, but magnetic vibration may be produced from a number of causes that can develop in the normal operation of the machine.

Before attempting to diagnose a case of noisy operation, it is well to determine if the cause is mechanical or magnetic. This may be done by disconnecting the motor from its load and then connecting it to the line. After it comes up to speed and is operating unusually noisy, if the switch is opened and the motor continues to vibrate, it will be known that mechanical causes are producing the noise. On the other hand, if when the switch is opened the noise immediately disappears, it will be known that the presence of current in the windings is the cause of the trouble.

Worn bearings or the rotor loose on the shaft may cause noisy operation of the motor. If the bearings become worn so as to allow the rotor to get too close to the stator core, the motor may fail to start and develop a very heavy vibration. If the motor is allowed to remain connected to the line under this condition, the vibration will be likely to injure the windings, either owing to excessive heating or vibration or both. With a squirrel-cage rotor, loose connections between some of the bars and end rings will cause unequal distribution of the current in the rotor and are likely

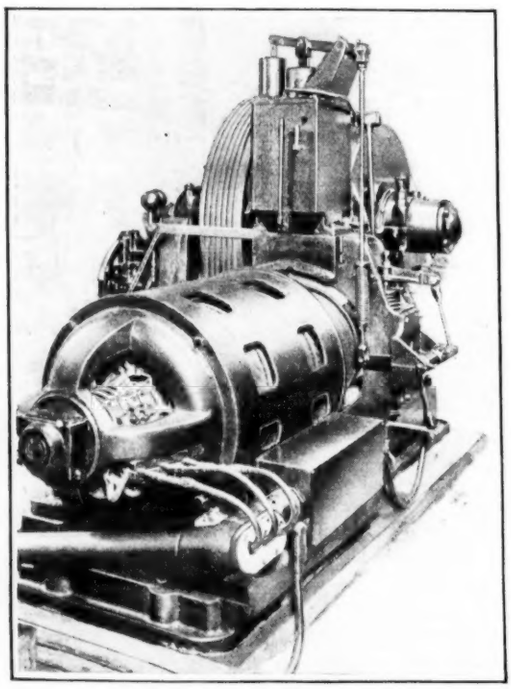


Fig. 5—Geared-traction elevator machine driven by double motor having a 1 to 3 speed range

to produce noisy operation. Vibration from this cause generally disappears when the motor comes up to speed. With a wound-rotor motor that had a number of grounds in the rotor winding, noisy operation has been known to develop.

Other causes of noisy operation are: Voltage too high; voltage unbalanced; open-circuits in part of the stator windings; short circuits in part of stator windings, which will generally be accompanied by excessive heating; and single-phase operation. The last is generally due to causes outside the motor, such as an open phase in the external wiring.

A polyphase motor cannot start single-phase, but will run single-phase if once brought up to speed and not too heavily loaded. Where a motor is started from reduced-voltage taps on an auto-transformer and

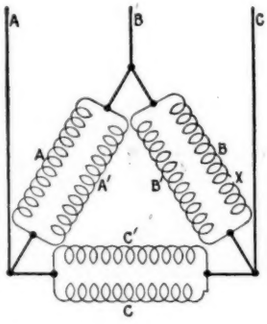


Fig. 6—Diagram of two - parallel delta-connected winding

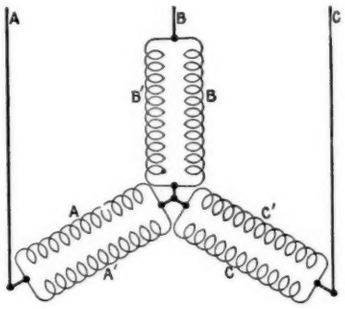


Fig. 7—Diagram of two-parallel star-connected winding

thrown to full voltage after coming up to speed, it is possible to have an open-circuit in one phase of the full-voltage wiring and have the connections complete on reduced voltage. Under these conditions the motor would start polyphase and would operate single-phase during the running period. This, however, will generally be accompanied by excessive heating of the winding, noisy operation, reduced speed, and if the motor is heavily loaded it may stop after being thrown onto full voltage, in which case the fuses or circuit breaker will generally open.

Failure of Motor to Start

Failure of an alternating-current motor to start when the load is free to move, may be caused by interruption of the power supply due to opening of the protective devices or otherwise, or to an opening in one phase of the wiring. If the power supply is completely cut off from the motor, when the starting switch is closed there will be no humming noise emitted by the motor. If only one phase is open, the motor will not start but will develop a loud roaring noise. Worn bearings that will allow the rotor to come in contact with the stator will cause the motor not to start; this fault will be accompanied by a loud roaring noise when power is applied to the stator. In this case, sometimes the motor may start while at other times it may not.

If a wound-rotor motor is connected to the line with the rotor resistance cut out, the motor is very likely not to start and will take a current from the line that will open the protective device. Opens in the resistance of wound-rotor motors will cause it to fail to start, but this condition will be accompanied by a loud roaring noise. A similar condition would exist with a squirrel-cage type motor in case a number of the bars are broken from the end rings. Grounds in the stator windings or short-circuits between phases may cause failure to start, although the conditions will generally cause the protective device to open.

Low torque may be due to low voltage, poor connection between the bars and end rings of squirrel-cage

motors, open circuits, short circuits and grounds in the rotor winding of wound-rotor motors. Open-circuits, short-circuits or grounds in the stator windings may cause low torque in addition to the other effects already referred to. If one phase is completely open, then the motor will not start. Many motors are wound with multiple circuit windings. For example, each phase has two windings in parallel, as in Figs. 6 and 7. If one of the windings were open, as *X*, Fig. 6, there would be two groups of windings in parallel between leads *A* and *B*, also between leads *A* and *C*, but only one winding between leads *B* and C. Such a condition in the windings would reduce the torque and cause the motor to take an unbalanced current from the line. Ammeters connected in the circuit would show the current unbalanced, but this should not be taken as an indication of a fault in the windings, since unbalanced line voltage would have a similar effect. If the motor is started cold and allowed to run for a short time, the phase with only one winding in circuit will heat more than the others. However, after it has been decided that the cause of the trouble is not external, the surest test is to break the windings into their parallel groups and test through them with a lamp.

Testing for Faults in Motor Windings

If low-voltage direct-current is available, the windings may be excited from this source and readings taken on the coil terminals with a millivoltmeter. If a section of winding is open-circuited, no reading will be obtained on the coils until the open is bridged by the meter, when a heavy deflection of the needle will occur. In making this test care should be exercised to take readings on coils only and not between phase, or misleading results may be obtained.

Another thing, with a delta-connected winding, the whole winding will be excited by connecting any two leads to the power source, where with a star-connected winding, Fig. 7, only two phases will be excited. In the delta-connection one phase will carry about double the current that the other two will, consequently the readings obtained on all three windings will not be equal. The winding carrying the larger current will give the greatest deflection on the millivoltmeter. To connect the voltmeter to the coil terminals, sharp steel points can be used as test lead terminals to pierce the insulation and make contact with the coils without removing the insulation.

Causes of Low Speed

Low speed may be due to low frequency, low voltage, rotor rubbing on the stator, poor connections or open-circuits in the rotor windings, open-circuits or short-circuits in the stator windings; the latter will generally be accompanied by excessive heating of the windings.

Heating of the windings may also be caused by high voltage, open-circuit in part of one phase, as previously explained, and the rotor rubbing on the stator. In elevator service overloads will rarely be the cause of heating.

Sparking at the slip rings of a wound-rotor motor is generally due to flat spots on the rings. These flat spots may be due to hard and soft spots in cast-iron rings or spongy spots in bronze rings; vibration that causes the brushes to break contact with the ring; brushes making poor contact or sticking in the holders; chattering of the brushes; and distortion of the rings at operating temperatures. However, in elevator service sparking at the slip rings of the motor is not generally a very serious trouble.

With alternating-current motors, if a phase is reversed accidentally on the power system, it will cause the motor to be reversed. This is a very dangerous condition and should be protected against with reverse-phase relays. Such relays will generally also protect against single-phase operation and should be installed on every polyphase alternating-current motor used on elevator service. Reversal of phases on alternating-current power systems has been the cause of wrecking elevator machinery.

There are other troubles that can develop owing to conditions in the controlling equipment, but these will be made the subject of another article.

Sand Blast To Clean Boiler Surfaces*

The Consolidated Gas, Electric Light & Power Company of Baltimore has used a sand blast rather extentively to clean metal surfaces prior to the application of protective coatings. The apparent ease with which the sand blast removed rust and other extraneous substances led to the belief that it might be profitably employed for the removal of clinkers from the lower rows of boiler tubes, as well as from the radiant-heat superheaters.

Slag having an average thickness of 2½ to 3 in. was removed from the surface of the superheater in 8 minutes, requiring the use of approximately 60 lb. of sand. The surface was thoroughly cleaned with no apparent erosion of metal from the surface of the superheater elements.

The sand blast was used, on the average, for 9 minutes on the lower rows of tubes, during which time approximately 70 lb. of sand was used, which resulted in all the clinker being cleaned off. There was, however, unmistakable evidence of erosion of the tube surface at isolated spots where the sand had been in contact with the tube for more than a fraction of a second, there being every indication that the protracted use of sand for removing clinker from tubes would result in a considerable decrease in tube thickness.

In order to obtain some figure by which to measure the amount of damage that might be done, a short section of 4-in. No. 8 gage tube was set up and the sand blast allowed to play upon it in one spot for 6 minutes, during which time approximately 60 lb. of sand was blown toward the tube, at the end of which time a hole approximately the size of a quarter of a dollar had been driven through the tube.

The result of these experiments would seem to indicate that the sand blast is effective in the removal of clinker from metal surfaces, and that where these surfaces are reasonably flat and the sand blast is applied in approximately the same plane as the surface, only very slight, if any, erosion will occur. The use of the sand blast, however, for removing clinker from boiler tubes, while effective, results in the erosion of the tube surface to such an extent that the protracted use of this process will probably cause extensive tube deterioration, for which reason it is felt that it should not be resorted to except under extreme conditions, and then only in the hands of an expert.

*From serial report of Prime Movers Committee, N.E.L.A.

Tests Made on Largest Hydraulic Turbine at Holtwood

ALLEN salt-velocity method used to test 20,000-hp. 54-ft. head turbine, connected to a short penstock on a river where the water has high electrical conductivity, offered many interesting problems in the successful application of this method of measuring the flow of water.

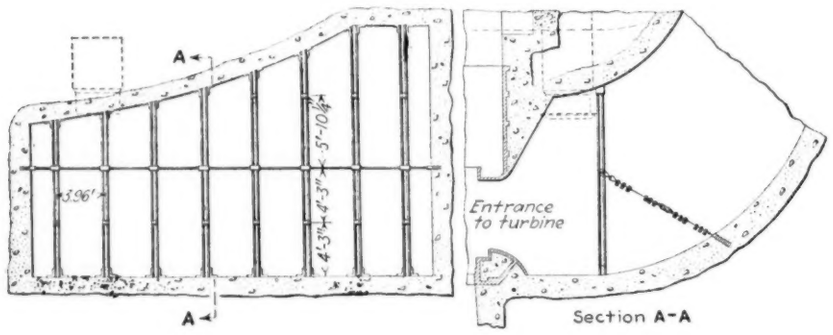
By ROBERT E. TURNER
Test Department, Pennsylvania Water and Power Company

IN THE installation of the last two units, Nos. 9 and 10, in the Holtwood development,[1] of the Pennsylvania Water & Power Co., it was desired that the maximum possible output and efficiency be obtained for the existing conditions. Runners of higher specific speed than those in the old machine were adopted and the two old settings, originally constructed for double-runner 13,000-hp. turbines, were converted into ones that would accommodate single-runner units rated at 20,000 hp. each, under a 62-ft. net head at a speed of 94.7 r.p.m. The runners, which are 156 in. in diameter, are set in concrete penstocks and spiral casings and equipped with Moody spreading-type draft tubes with spiral collectors. A section elevation through one of these turbines is shown in Fig. 3.

The general problem in the tests made on unit No. 10 was that of measuring the hydraulic input and the horsepower output delivered to the generator shaft. The measurements of the hydraulic input required the measurement of the head and the quantity of water used by the unit. The output was obtained by measuring the energy generated electrically, to which were added the separate losses of the generator determined previously from generator retardation tests.

The net or effective head used in calculating results was the water elevation at the entrance section of the turbine casing measured by means of piezometer plates, plus the velocity head at this section, less the sum of the elevation of the water surface and velocity head at the discharge section of the draft tube.

It is the determination of the quantity of water used by the unit, or the discharge, which is the most difficult and most interesting of all measurements in tests of turbines at site, especially in low-head plants. For this test the salt-velocity method devised by Prof. C. M. Allen, of Worcester Polytechnic Institute, was used to measure discharge. The test was personally conducted by Professor Allen with the co-operation of the test department of the Pennsylvania Water & Power Co. and witnessed by a representative of the turbine manufacturers. For a description of this method and its development see a paper by C. M. Allen and E. A. Taylor in the transactions of the A. S. M. E., Vol. 45, 1923, and *Power*, April 29, 1924. This method of water measurement is based on the fact that salt in solution increases the conductivity of water. The method consists simply of introducing the salt solution at the upper end of the penstock by means of quick-acting pop valves near the headgates and recording graphically its passage through a group of electrodes at the lower end of the penstock.



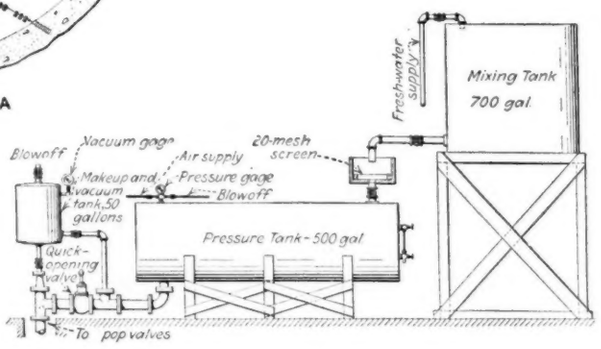


Fig. 1 (Left)—Eight electrodes were installed at lower end of the penstock near the entrance to the turbine. Fig. 2 (Below)—Equipment used to mix and supply the salt solution to the 40 quick-acting pop valves in the penstock entrance

The passage of the salt solution from the pop valves to the electrodes was accurately timed in these tests. As the volume of the penstock between the same points had been accurately determined by previous calculations from an actual survey, it was possible to obtain the discharge in cubic feet per second by dividing the volume in cubic feet by the time in seconds. While this method has been approved and has become one of the standard methods under the testing codes for hydraulic turbines, nevertheless, its application to the particular test was unique in many ways and the difficulties encountered and overcome served as a further proof of its usefulness and increased confidence in its application.

Salt solution was injected through a system of forty quick-acting pop valves located just back of the head-

[1] Described in *Power*, Sept. 30, 1924.

gates. The pop valves were fed by a four-inch pipe leading from the pressure apparatus in the gatehouse down through the vent pipe to pipework in the penstock. Fig. 4 shows the location and details of the pop valves and piping. The necessary pressure and mixing tanks, pipes, valves, etc., were located in the headgate house. Fig. 2 shows the arrangement of this apparatus. A quick-acting valve was used to control the brine injection and a contact switch on this valve handle was wired to the recording ammeter and thus recorded the injection of the salt charge on the recording chart.

Eight pairs of vertical steel electrodes were equally spaced across the electrode section, located at the lower end of the penstock approximately 45 ft. below the pop valves, each electrode being made up of two 4x½-in. steel plates separated 4½ in. with busbar insulators and running the full height of the penstock (see Fig. 1). The electrodes were all wired to a terminal board at the recording apparatus, where in turn any combination could be wired to the recording ammeter.

By means of a magnetic relay on a calibrated seconds pendulum the time in seconds was recorded on the chart by a jump-spark arrangement which burned a small hole in the graphic record every second. A reproduction of a typical test record chart is shown in Fig. 5. The line of dots near the top of the chart indicates the time, the distance between dots representing one second. The square top diagrams *A* represent the introduction of brine or the opening of the quick-acting valve, while the curves *B* are the pen records of the passage of the salt solution across the electrodes with current through the recording ammeter. The time for

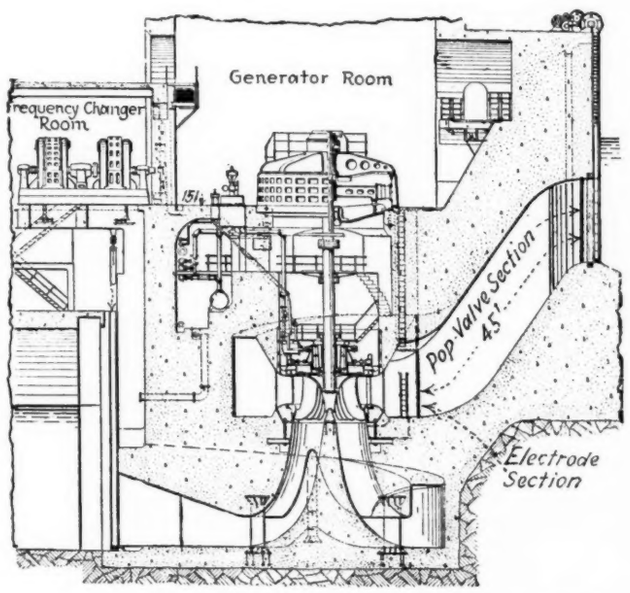


Fig. 3—Turbine tested has spreading-type draft tube with spiral collector

the passage of the salt solution through the penstock is the time between the mean time of the valve opening and the time of the passage of center of gravity of the salt charge across the electrodes. The center of gravity of the salt charge is the center of gravity of the area inclosed by the curve of current against time. The time, represented by the distance between the mean time of the pop-valve opening and the center of gravity of the curve obtained from the electrodes, is the average time it takes the water to travel the distance from the pop valves to the electrodes. Therefore, dividing the volume of the penstock between the pop-valve openings and the electrode center-line by this time in seconds gives the discharge of the unit in cubic feet per second.

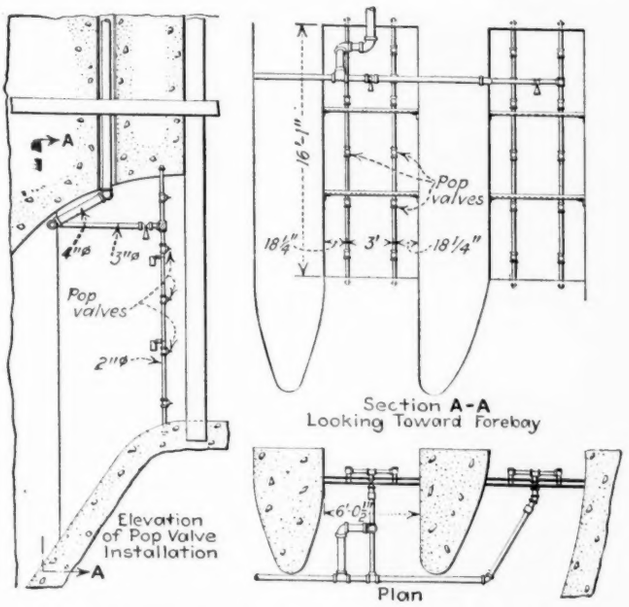


Fig. 4—Arrangement of pop valves in two of the four openings from the forebay to the penstock

The difficulties encountered in applying the salt-velocity method to this test were, briefly, the type of setting as previously described, the high conductivity of the Susquehanna River water, and the length and size of pipework necessary from the pressure system to the pop valves. Since the penstock was so short, approximately 45 ft., the time of the passage of the salt solution from the pop valves to electrodes was short, less than 5 sec. in some cases. It was necessary, therefore, to make check tests and take special care in the determination of the time.

In order to make an accurate determination of the passage of the center of gravity of the salt charge across the electrodes, it is necessary that the deflection of the ammeter be large enough to make the area inclosed by the curve of current against time great enough that its center of gravity may be accurately determined. The Susquehanna River flows through developed coal fields and manufacturing districts, and the water has a relatively high conductivity due to the mine wash and manufacturing-plant refuse. This condition, naturally, is more marked in periods of low flow since the quantity of impurities thus deposited in the river is nearly constant.

As this test was made during low-flow period, the high conductivity of the water made it difficult to obtain sufficient deflection on the recording ammeter in spite of the introduction of a large amount of brine, approximately 40 gal. per injection. The pop-valve openings were set at the maximum which the pressure tank could supply and still maintain sufficient pressure. The voltage on the electrodes was increased, making the ammeter deflection greater, but causing the curve to go off the chart. The curve was then brought to approximately the center of the chart by shifting the

zero resting point of the pen back considerably from normal. It was also found that the impedance of the recording ammeter was approximately 35 per cent of the total impedance of the circuit. To decrease the ammeter impedance, another impedance of approximately equal value was paralleled with the ammeter. With these arrangements it was possible to get a more desirable curve showing the passage of the salt solution across the electrodes.

Considerable trouble was experienced at first, due to

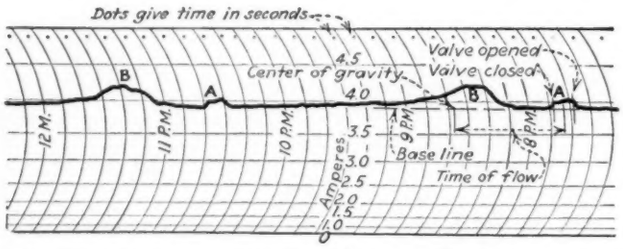


Fig. 5—Curve A, indicates the period when the pop valves were open and B when the salt solution passed the electrodes

entrained air forming in pockets in the supply pipes to the pop valves and causing inaccurate operation of the pop valves. In this particular installation extraordinarily long and large supply piping was needed from the headgate house pressure system to the pop valves. Entrained air in the salt solution, therefore, presented a much greater problem. In fact, owing to this difficulty it was impossible to obtain consistent results. To draw this entrained air from the pipes, Professor Allen conceived the idea of installing a makeup and vacuum tank on the discharge side of the quick-acting valve

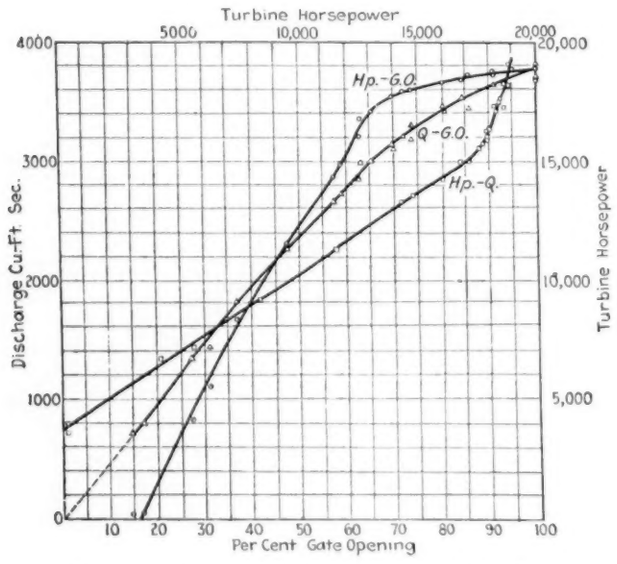


Fig. 6—Characteristic curves of the turbine at 54-ft. net head

(See Fig. 2). This tank was filled with brine before each run of tests. After each operation of the quick-acting valve, the supply line from the makeup tank was opened to allow salt solution from the tank to fill the piping, thereby causing a reduced pressure in the makeup tank drawing the entrained air from the supply pipes. With this arrangement it was possible to get consistent results for the water discharge through the turbine.

The fact that these difficulties were overcome increases the usefulness of this method for the measurement of discharge of turbines tested at site and increases confidence in the results of such tests.

Tests were made at various heads and gate openings. All together 50 tests were made, but some of these were rejected in final computations for various reasons which developed during tests. Several tests were made at some of the same gate openings. In each of these tests ten brine injections were made during which time the measurements of generator output, head, gate openings and speed were made, the gate opening being held constant during each run with the turbine on hand control.

For the final curve of efficiency at 54 ft. net head, which is the head used for the guarantee, the corrected test values of discharge and horsepower were plotted against true gate opening in per cent and the discharge

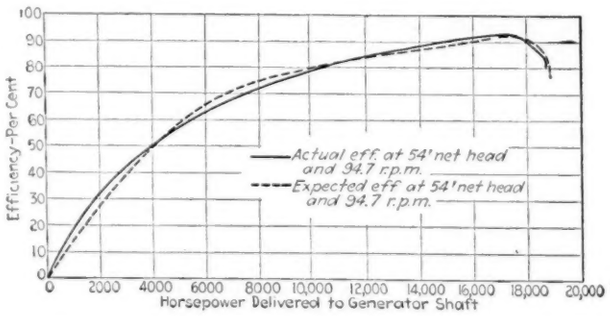
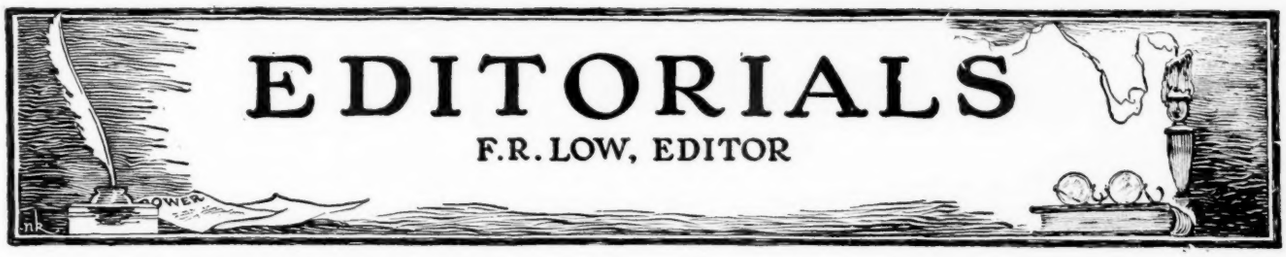


Fig. 7—Actual and expected efficiency curves of the turbine at 54-ft. net head

plotted against horsepower, Fig. 6. These curves were balanced one against the other to determine the best location for each curve to correspond to test points; that is, points on the curve of discharge against horsepower correspond to the horsepower and discharge for a certain value of gate opening on the other two curves. Then, from the values taken from the discharge-horsepower curve, the efficiency was computed and plotted on a curve irrespective of test results of efficiency, Fig. 7. Test results were then plotted and conform very closely to the curve, making a double check on the final efficiency curve. The dotted curve shows the expected efficiency curve as predicted by the manufacturers from tests on a homologous runner. Note how closely the actual efficiency curve checks the expected efficiency curve.

The results of these tests show a maximum efficiency at 54 ft. net head of 92.8 per cent at an output of 17,200 hp. with a gate opening of 65.5 per cent, which is very good. However, due to the higher specific speed of this runner necessitated by the desire of maximum possible power output with the converted setting, as previously explained, the efficiency curve has quite a peaked characteristic and it is evident that it would be impossible to operate at anywhere near the maximum efficiency over a wide range of output. In addition, although the turbine runner itself has a maximum efficiency of 92.8 per cent at 54-ft. net head, under a 54-ft. gross head the turbine, due to site losses, is approximately 4 per cent less efficient. After all, this efficiency of the turbine at site is the efficiency of interest to any power development.



Solving the Dust Problem

THE ash and coal dust discharged from pulverized-fuel boiler plants has perhaps been given more publicity than it merits. Such publicity, however, merely increases the difficulty of the problem and emphasizes the importance of speedy and satisfactory solution. Each type of dust seems to present its own peculiar problem, and it is possible that developments will lead to the use of distinct apparatus for the two jobs.

The problem of intercepting ash dust in the power plant is not made easier by the fact that the ash dust has thus far been without commercial value. Apparatus for preventing its escape to the atmosphere simply recovers something that must be removed from the plant at further expense. Of course, means may be developed whereby the intercepted ash dust will be used for definite commercial purposes. Two or three tentative efforts in this direction are now on foot.

Ease of inspection and maintenance are important considerations, and it is to be hoped that designers will work out a sectional construction, with arrangements for removing any section from the breeching without stopping the flow of gas and, so far as possible, without interfering with the functioning of the remaining sections of the dust catcher.

Attention has recently been directed to the Cottrell electrostatic precipitator, used in the cement and chemical industries. One large central station has installed an extensive equipment, and in another station experimental installations are being studied with a view to the development of a design that shall be more reliable as well as more effective.

The performance of an electrostatic precipitator is a matter of maintaining corona around a set of wire electrodes. The presence of an excessive content of conducting matter in the dust is likely to cause arcing, which interferes with operation, but the ash emitted from well-controlled furnaces will carry relatively little unburned carbon—probably less than 20 per cent. The success of the Cottrell precipitator in the cement plant seems to augur well for its performance in the boiler room, since cement and ash dust are closely similar. More or less development work must nevertheless be done, and it is of interest to every power engineer to know that such experimental studies are in progress, with reasonable hope that before many months there will have been developed a design of precipitator that will be entirely practicable for installation in the boiler room and effective in removing most of the dust carried by the outgoing flue gas.

The problem of reducing the quantity of coal dust discharged from pulverizing plants, on the contrary, is of a different nature. The coal dust, with its high carbon content, is a good enough electrical conductor to make it very difficult, if not impossible, to operate an electrostatic precipitator successfully. It has long been recognized that water-spray washers are not altogether effective in removing fine coal dust because of the difficulty of wetting it. This is a problem to which every engineer interested in pulverized-coal development must address himself, and already there are a number who are giving attention to it. While there are no definitely favorable indications at present, the fact that these capable men are working hard on the problem gives ground for the hope that presently there will be a solution for this difficulty as well.

Information for the Operator

THE power-plant operator needs information from the manufacturer of equipment. It often proves difficult for the operating engineer to determine the internal construction of some of his machinery without tearing it apart, and this he is rarely able to do because he must maintain service. Without information the operator is often at a loss to understand and correct abnormalities in operation. His efforts, far from being helpful, may be positively detrimental.

The operator is not primarily concerned with how things are made, or with exactly what materials are used. Except in special cases he does not need dimensioned working drawings sufficient for the reproduction of a machine or a part of it. No one should expect, or can fairly ask, a manufacturer to furnish such drawings, save in special cases.

The operator's need is for clear assembly drawings, with occasional subassembly details, to tell him what is inside the casing of his equipment and how it functions. This will help him, in case of breakdown, to discover more speedily what is the matter and to restore service quicker and more effectively. It should be noted in passing that the operator is not interested in drawings showing how equipment was made last year. Few things are so exasperating as obsolete drawings.

The matter goes further. Why should a manufacturer not tell the operator a few of the troubles to which his equipment is especially liable? This involves no suggestion of defect in design or manufacture. Experience and common sense indicate that every piece of equipment is better adapted to certain uses than to others, and has definite operating characteristics. There are things that must be done and things that must be avoided. The operator should know about certain symptoms of trouble without waiting for bitter experience.

The manufacturer has little to lose by frankly recognizing the limitations of his product, for while frankness may result in some restriction of sales, is it not true that such limitation will help the manufacturer more than it will hinder him? It is of no value to a manufacturer, in the long run, to sell his equipment for purposes to which it is ill-adapted. The user, discovering equipment to be unsuitable for the purpose for

which he purchased it, often concludes that the manufacturer is either incompetent or dishonest, and such an attitude on the part of a purchaser does little to help the seller's business; indeed, great harm may be done by people who are determined to oppose purchasing anything whatever from a manufacturer who has sold them one small piece of equipment that proved unsuitable for the task to which it was applied.

Full and frank co-operation will enable the operator to secure more satisfactory service from his equipment and will give the manufacturer an invaluable source of intimate information to guide him in improving his product. The operator will be less inclined to emphasize the deficiencies of the apparatus and more likely to be an enthusiastic booster for the manufacturer.

Simplified Practice In the Power Plant

WHEN an industry passes through its development stages, opinions as to what is the best procedure are generally not very well crystallized and as a result a wide diversity of practice comes into use. This diversity in practice and in shapes and sizes creates one of the heavy burdens industry is laboring under today. To assist in eliminating this burden as far as possible, the Division of Simplified Practice was created in the Department of Commerce by Secretary Herbert Hoover.

Simplified practice should not be confused with standardization, although the former may lead to the latter. Simplified practice simply means that the interested parties in a certain industry get together and after a careful survey agree to eliminate, so far as possible, from their trade, shapes, sizes and forms for which there is little demand or for which a more standardized product can be used. It is estimated that simplified practice as adopted by about forty-five industries in this country, has resulted in a saving of about six hundred million dollars in the last five years. The reductions in varieties have ranged from a maximum of ninety-eight per cent to about thirty per cent. As an example, before simplified practice was adopted in the brick industry there were forty-four different sizes of common bricks; this has been reduced to one size.

Although some of the simplified practices already adopted reach into the power plant, there is a tremendous opportunity for further simplifications in this field. Recently, attention was called in these columns to the wide diversity in carbon-brush sizes, a large percentage of which could undoubtedly be eliminated by common consent of those interested in this matter getting together and making a survey of the situation. Fortunately, the initial step has already been taken in this direction.

Switchboard slates offer a large field for simplified practice. If nothing more than a nomenclature was agreed upon, material benefits would result. As it is now a manufacturer is not sure of himself when putting a bevel on a piece of slate without having a drawing showing what is meant by the dimensions of the bevel. In some specifications the length of the bevel may mean the long side of the triangle, where in others it refers to the length of the legs of the triangle. Why some standard for such a simple dimension as this has not been adopted long ago is difficult to understand. Already simplified practice has been applied to blackboard slate and has resulted in a reduction in the number of sizes from 251 to 25; this indicates the possibilities with switchboard slate.

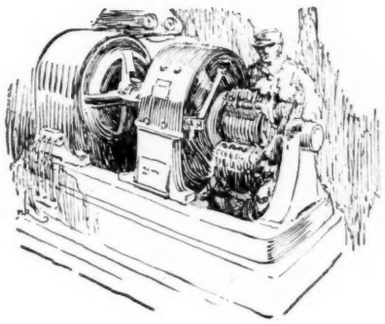
Packing sizes have been determined largely by the whim of the engineers who designed the machines, and the packing manufacturers have made their product to suit the machine's dimensions. The space for packing does not require adopting a wide range of dimensions. If seven-sixteenths is sufficient space, the dimension can just as well, and probably better, be made one-half inch and this size of packing used. Many other materials used in the power plant might be cited for simplified practice, but the foregoing are sufficient to show the possibilities. This movement is of interest to equipment and material manufacturers, and the users all have a vital interest in the problem and should lend their hearty co-operation to any effort to simplify their practices.

In a recent statement to the press, Thomas L. Chadbourne, President of the American Association for Labor Legislation, places the blame for some of our coal troubles on the neglect to publish the United States Coal Commission report. "Three years ago," he contends, "the public demanded to know about the coal industry, and Congress provided $600,000 to have the Fact Finding Commission dig them out. Two full years have now elapsed since the Commission finished its work, yet the findings have not been made available to the public in printed form, although authorization for the printing has been made by Congress." Mr. Chadbourne appears to pass over the fact that the report in multigraphed form, exclusive of some supporting data, has been available to Congress and all agencies that might be in a position to initiate or influence action in solving the problems in the anthracite industry, and numerous summaries of the report appeared in the public press at the time. One of the legitimate criticisms that attaches to this report is its voluminous character and the absence of any definite summation of the findings such as could be readily digested by the public. In view of this it is doubtful if the public would be much better off if the report were printed and broadcasted.

Will the next swing be to higher steam temperatures? In a recent Prime Movers' Committee (N.E.L.A.) report A. G. Christie and D. C. Turnbull, Jr., predict the early development of materials suitable for much higher temperatures than the present upper limit of 750 degrees and suggest that high temperatures with moderate pressures (around 400 pounds) may show up better than extreme pressures (around 1,000 pounds) at temperatures below 750 degrees. In this connection the possibility of combining extremely high pressures with the highest permissible temperatures should not be overlooked. This combination might eliminate the necessity for reheating for pressures above 500 pounds as is now done in several stations.

The biggest commercial obstacle to be overcome in the development of the Bay of Fundy Power at Eastport, Maine, is the absence of any near-by market for the power. Maine and near-by sections of Canada could (for the present at least) use but a small fraction of the half-million horsepower that would be made available by the construction of the projected dam.

Practical Ideas from Practical Men



WITH a view to stimulating engineers into the habit of recording for the benefit of brother engineers, unusual occurrences, how these were met and other practical expedients adopted in the operation of their plants, Power has decided to award two cash prizes each month during 1925. One of $25 for the best and another of $15 for the second best practical letter on plant operation or practical kinks received during the month. This is in addition to payment for the contribution at space rates. The winners for September will be announced next month.

Shrinking a Long Sleeve on a Shaft

There came into the shop one day a job that set the foreman and master mechanic thinking. They had to put a bronze sleeve 20 ft. 6 in. long on a shaft 28 ft. in length. It was a job of shrinking. The shaft was 14 in. in diameter and the sleeve was 15 in. in diameter outside and 0.010 in. less than 14 in. inside. The designers had made provision for the placement of the sleeve by making a ½-in. counterbore in it and a corresponding step on the shaft, for half the length. This was done to prevent dragging in the event of the sleeve warping, ever so slightly, during heating.

Obviously, the job had to be done quickly. The shaft could not lie horizontally on any sort of a support, so it had to be placed on end and the sleeve lowered over it. A little joker appeared right there. When the sleeve was suspended above the shaft, the total height would be something more than 48 ft. 6 in. and the height of the shop cranes was just under 40 ft. Two cranes were employed and the sleeve was lifted between them, since there was plenty of headroom.

The rigging was done first. Then the shaft, coupling end down, was stood on the floor of the shop and slings were attached to the sleeve so that it would be picked up perfectly plumb. A gas jet was set up in the floor and supports for the sleeve placed around it with adjustable draft doors between them. All the doors in the shop were closed to keep out drafts of air. The sleeve was set up like a chimney over the gas jet, and a damper was placed on the top to restrict the flow of heat and bring about an even distribution.

Then we lighted up and checked up on the rate of expansion, adjusting the draft at the bottom of the sleeve and the damper at the top until an even distribution was obtained. The temperature of the sleeve was not checked; instead, it was measured for expansion with micrometers. At the end of an hour and thirty minutes it had increased in size ¼ in. (0.0156), making a clearance of 0.0056 in. total, for, as previously mentioned, the sleeve was 0.010 in. small to make a tight job of shrinking. Finally, the sleeve was lifted and lowered into place without trouble. Everyone who understood what was at stake was quite nervous, however, thinking of what the result would have been had the sleeve been warped more than five-thousandths. If it had tightened on the shaft before it was in place, the heat would have run directly into the shaft and a seizure would have meant cutting the sleeve to get it off, since a second heating would expand both parts.

The object in placing a solid sleeve on the shaft in place of jointing it in the middle, is in itself interesting. It was the propeller shaft for a ship, and the sleeve was placed on it as a protection against the action of sea water.

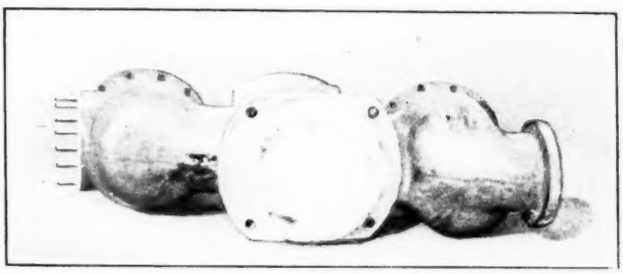
Steel and bronze together in the presence of salt water set up a galvanic action that eats away the steel. The owners of this ship were afraid that a joint in the sleeve would leak and an attack on the steel would follow and eventually weaken the shaft to such an extent that it would break, or the electrolysis would find its way between the sleeve and shaft and loosen the former, both of which have taken place. A. B. NEWELL.

Woodbury, N. J.

Emergency Repair of Pump Casting by Cast-Iron Brazing

Some time ago the crosshead broke on a 13x15-in. triplex pump that is used for pumping coal tar, causing the piston to punch the bottom out of the cylinder. The section broken out was in small pieces, so it was not considered advisable to save them as means of making the repair.

It was necessary to get the pump back in service again as quickly as possible, and it was up to the mechanical



Pump was repaired by brazing a steel-plate patch in the bottom of the cast-iron cylinder

department to make the repair. It was finally decided to cut out a piece of ½-in. boiler plate with an acetylene cutting torch to make a patch, the patch to be dished and cut to fit the irregular opening. The next question was whether to make a weld between the cast-iron and steel or braze these metals with bronze rod and flux.

Welding of steel and cast iron is accomplished with cast-iron filler rod and flux. The weld, however, is liable to be brittle and porous. To braze a joint of this kind it is only necessary to bring the metal to a dull red

heat before applying the bronze, and added to this is the advantage that the bronze is ductile.

Brazing of cast iron is a relatively new application of the welding blowpipe, and although we had never attempted anything the size of the pump, the master mechanic decided to try it.

The surface of the metal in the break and on each side for about 1½ in. was polished bright with a portable grinding wheel before the bronze was applied. The brazing operation required two hours, and 8 lb. of bronze was applied. The job was satisfactory, and the pump was back in operation in a short time.

St. Louis, Mo. P. A. SPAULDING.

Plant Upkeep and Labor Costs

Of the many difficulties to be faced by the plant engineer, the problem of costs is one that is continually coming up, particularly in textile works where the power plant is looked upon as "an expensive nuisance." The engineer occupying a position of this kind does not get much rest or satisfaction. Every item of new equipment or renewals has to be "begged for." Courts-martial are held over all invoices before being passed for payment, and any extra cost for wages due to breakdowns or accidents is sure to produce more criticism for the poor engineer. Much more could be added to the list of troubles, but many engineers already know them and some no doubt would welcome a solution to the problem. Unless one is thoroughly acquainted with the exact conditions it is not possible to offer any specified cure.

A method applied by a friend of mine might be given, as it proved successful and completely cured his troubles. This friend suffered all the troubles just enumerated and was further handicapped by having two old boilers that were worn to a highly dangerous condition. He had reported the condition of the boilers to the directors repeatedly, but without avail. Finally, in desperation, he served upon the Chairman of Directors a written notice describing the actual condition of the boilers and renouncing all responsibility on his part from that date. He also intimated that a copy of this document was being sent to the chief of police and the government inspector. This action resulted in new boilers being installed without delay. Since that time he has had little trouble in getting his recommendations considered.

The question of labor costs, particularly as applied to overtime, must also be dealt with from individual knowledge of each case. No doubt many breakdowns should not occur, but with a staff that is being paid (as many firms insist upon) upon an hourly rate with time and a half and double time for overtime, it is contrary to human nature to expect the workmen to take special pains to prevent extra work.

Two years ago the chief engineer of a large local works, after a great deal of trouble, obtained permission to place his staff of about twelve men, including fitters, machinists and millwrights, upon a fixed weekly salary. This he based on a 20 per cent reduction on the total average wages paid over the previous five years. In addition he offered full salary up to four weeks' sick leave and fourteen days' holiday a year. This offer was accepted with much pleasure by his staff and the results have been most surprising. All the men soon became expert trouble detectors, and whereas the chief himself had previously to keep up continual vigilance on the whole plant to avoid breakdowns and overtime, his attention is now called, by the members of his staff, to possible trouble long enough before the breaking point is reached to permit renewals being made well in advance. He has also found that many items that previously occupied the men half the night are now slipped into place during a meal hour.

Although it was difficult to get this firm to adopt this system, it would not under any circumstances go back to the old method. And I am quite sure that the plant engineer, who is continually in trouble over extra hours and wages, can, by adopting similar methods, not only effect a considerable saving in labor cost, but make his own position a much happier one.

Belfast, Ireland. F. P. TERRY.

Using High-Pressure Steam in the Industrial Power Plant

There is under construction in the shop of a well-known American engine builder an engine that is designed to operate at 350 lb. pressure. It is a compound unit with the high-pressure cylinder exhausting at a back pressure of over 100 lb. into a standard Corliss-type low-pressure cylinder, which in turn may exhaust at a considerable back pressure to a manufacturing process or may run condensing. If the proper degree of superheat is used at this high pressure, it is evident that a low water rate can be obtained even if the engine is operated non-condensing. This suggests an idea that can be applied profitably to many industrial plants that have old engines operating at pressures around 100 to 125 pounds.

The average industrial plant seldom operates at higher pressures than this, consequently the water rates of their engines are too high for the production of cheap power. Also many industries operate their engines condensing and heat buildings with live steam for the reason that their engines are loaded to a point where they must run condensing. Even in plants where the engine could carry the load non-condensing, the water rate is so high that it would exhaust more steam than the heating system would require and in steel plants and some other metal-working industries the steam used for power production is often double that required for heating and practically no steam is needed for process work in these industries.

Consider, for instance, a single-cylinder condensing engine of 500 hp. running 24 hours a day with a water rate of 18 lb. per hp. at 105 lb. gage pressure and 25-in. vacuum. The steam required to heat buildings is around 6,000 lb. per hour maximum, so the total steam consumption is 15,000 lb. per hour. If the engine is operated at 3 lb. back pressure, the steam consumption is increased by about one-third, hence the engine will exhaust 12,000 lb. of steam per hour, or more than twice what is required by the heating system, and furthermore, if it was already overloaded when run condensing it would be impossible to operate non-condensing. But if 6,000 lb. of steam is generated at 350 lb. absolute and high superheat and a high-pressure engine of 250 hp. is installed to expand the steam from 350 lb. down to 105 lb. and exhaust into the old engine at that pressure, the total load can be carried on the steam required by the heating system. The water rate of the low-pressure engine operating non-condensing will be about 24 lb. per hp.-hr., and the high-pressure

engine must have a water rate of 24 lb. per hp. in order to carry the load of 500 hp., or 6,000 lb. of steam.

According to the steam table the temperature of steam at 350 lb. absolute and 250 deg. superheat is 682 deg. F., and this is probably the upper limit of temperature with an engine of this type. With a water rate of 24 lb. per hp. the high-pressure engine would extract 106 B.t.u. from each pound of steam. The total heat content of the initial steam is 1,349 B.t.u. per lb., and consequently the exhaust will be superheated about 99 deg. F., since it contains 1,243 B.t.u. per lb. and the steam table gives the heat content of steam at 105 lb. gage pressure and superheated 100 deg. F. as 1,244.1 B.t.u. Using the formula $U = H_s - H_2 - T_2 (N_s - N_2)$, we find that there are 126 B.t.u. between 350 lb. and 105 lb. available for converting into energy. To secure a water rate of 24 lb. per hp.-hr. under these conditions, demands a Rankine cycle ratio of 84 per cent, and obviously we can expect so good a ratio from this type of engine. But the exhaust at 105 lb. pressure is superheated 99 deg. and higher if the water rate of the high-pressure engine exceeds 24 lb., therefore, with a superheat in the region of 100 deg. entering the low-pressure engine we can expect a lower water rate than 24 lb. in this unit. The high-pressure engine could be designed to show a Rankine-cycle ratio of at least 80 per cent, and this would give a water rate of about 25.5 lb. for the high-pressure engine and about 22 lb. or a ratio of 77 per cent for the low-pressure engine.

In ordinary compound condensing engines the Rankine cycle ratio is around 65 per cent for the entire engine, being as low as 50 per cent for the low-pressure cylinder expanding steam from around 10 lb. gage down to 26 in. vacuum, and the high-pressure cylinder expanding from pressure around 125 down to 10 lb. often attains a Rankine-cycle ratio as high as 80 per cent. We can fairly assure that 80 per cent can be secured at 350 lb. pressure and 250 deg. superheat. An engine utilizing the steam at this pressure and temperature and exhausting into the present engine at 105 lb. pressure, should reduce the steam consumption by at least 60 per cent of what is now used in winter, and in summer a saving of at least 45 per cent is possible with the low-pressure engine running condensing.

Assuming that the high-pressure engine and boiler cost $150 per hp. to install and 15 per cent fixed charges are allowed, the fuel saving will be more than double the fixed charges. The steam flow in the old plant being 15,000 lb. in winter and 9,000 lb. in summer, gives an annual coal consumption of 5,400 tons, 300 days a year; at $7 a ton the fuel cost is $37,800, and nearly half of this is saved. This gives a fuel saving of $18,900 against fixed charges of $5,625, and the fuel saving about pays for the plant in two years.

In the case of plants requiring process steam at 20 lb. pressure and upward, 24 hours a day, the gain is much greater. Take the case of a plant having a process load of 20,000 lb. of steam an hour and requiring 15,000 hp. There are many plants of this type, and at present their power supply is usually furnished in one of the following ways: A compound-condensing engine with a water rate of at least 13 lb. of steam per hp.-hr., a bleeder turbine of 1,500-hp. with a water rate for the high-pressure end alone of 40 lb. per hp. taking steam at 150 lb. gage and exhausting or bleeding at 20 lb. and a water rate for the high- and low-pressure stages expanding steam to 28.5 in. of vacuum, of about 11.5 lb. per hp.-hr. Or current may be purchased at 1c. per kw.-hr. with a connected load of 1,500 hp. and low-pressure steam generated for process work. Taking the first plant, the total load is around 40,000 lb. an hour. Seldom is any attempt made to bleed the receivers of compound engines except for small amounts, and if tried in this case, only a fraction of the necessary process steam could be bled without reducing the capacity of the engine. Even with the bleeder turbine, assuming that 20,000 lb. of steam per hour is bled, the total steam flow will be at least 33,000 lb. per hour since the 20,000 lb. of bled steam generated only 500 hp. in the high-pressure end of the turbine and 11,500 lb. of additional steam must be supplied to the turbine to carry the load. With purchased current the cost of 1c. per kw.-hr. for 1,500 hp. 300 days per year is $81,000, and the process steam will require at least 1.1 tons of coal per hour, or 7,920 tons a year, at a cost of about $55,440, making the total cost of steam and power $136,440 in this plant. In the turbine plant the annual fuel cost, assuming an evaporation of 8 lb. of water per pound of coal, will be $\frac{33{,}000 \times 7{,}200 \times \$7}{8 \times 2{,}000} = \$103{,}950$ exclusive of labor and fixed charges, and the first plant will have cost figures nearly 20 per cent higher than the turbine plant, which gives a fuel cost of over $126,000 a year.

Now if 20,000 lb. of steam, which represents the minimum amount that will run the plant, is generated at 350 lb. pressure and superheated, it can develop 1,500 hp. if expanded in a high-pressure engine to 125 lb. back pressure and then expanded in the old engine from 125 lb. to 20 lb. and exhausted to the process. In the case of the compound condensing engine the old low-pressure cylinder could be discarded and the high-pressure cylinder serve as a low-pressure cylinder.

This gives a reduction in fuel costs of practically 50 per cent in the first-named plant, or about $63,000 per year and reductions in labor cost of operating the boiler room.

The saving in the second plant using a bleeder turbine would be slightly less, since the high-pressure end of the turbine expanding steam from 150 lb. to 20 lb. is not as efficient as the high-pressure cylinder of a compound engine between the same pressures. If the high-pressure engine sends superheated exhaust to the turbine, the efficiency is improved somewhat, but the water rate would be at least 15 per cent higher than the water rate possible in the first plant, and about 2,000 lb. of steam per hour must be expanded to the condenser pressure, since the total load of 1,500 hp. cannot be carried on 20,000 lb. in this plant using the turbine as the low-pressure part of the plant. The amount of steam going to the condenser could be greatly reduced by using a high-pressure feed-water heater and bleeding steam at the exhaust connection between the high-pressure engine and turbine. The fuel saving in plants of this type will be from one-third to one-half.

In many old plants equipped with bleeder turbines of 1,500 hp., it is impossible to bleed over 10,000 lb. of steam per hour without reducing the turbine output, hence the possible saving to be effected by the foregoing plan is much greater. In the case of the plant purchasing power and generating low-pressure steam the cost of current could be reduced from 1c. per kw.-hr. to about ½c. by installing a high-pressure plant and floating the power load on the process steam demand.

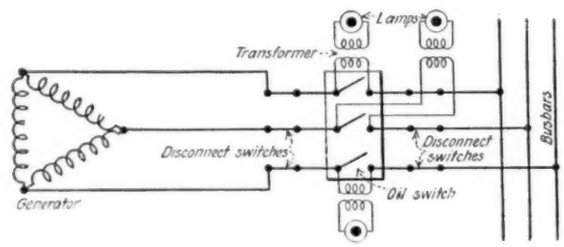
Brightwood, Mass. A. F. SHEEHAN.

Comments from Readers

Phasing Out Alternating-Current Generators

I have been reading the articles published from time to time on phasing out and synchronizing alternating-current generators. The process of phasing out and synchronizing alternating-current generators is quite simple if the machines are designed to run together and the correct instruments are at hand to do the work. These do not have to be elaborate in extent, as sometimes only a few lamps are used to do the phase out and synchronizing with. On machines of large size a synchroscope should be used for indicating synchronism. Phasing out and checking the machine's connections to make sure they are correct is essential, as serious damage may be done through error in connections that are assumed to be correct and have not been properly checked.

One instance comes to mind where a switchboard was being relocated. The synchroscope was connected to one of the old generator panels by extending the



Connection for phasing out an alternating current generator

wiring from the new board. After the old board was dismantled and the synchroscope wires cut loose, one of the machines was synchronized to the busbars, but when the switch was closed a heavy overload was thrown on the system. This led to the conclusion that the machine had its leads to the busbars interchanged. After the machine was shut down, the leads were interchanged and the man in charge tried to phase the generator to the busbars with two transformers. As the generator was a three-phase type, three transformers should have been used to show the correct phase relation. After checking the connections and finding them to appear to give correct phase relation, the machine was again synchronized with the busbars and the oil switch closed. In this case the overload on the machine was more serious than before. The operator then changed the generator leads again and put a piece of fuse wire in each of the leads between the oil switch and the busbars. When the machine was again synchronized, there was a loud roar and the busbar arced over, which was extinguished only by taking all the generators out of service.

The job was then done right by the maintenance man obtaining three transformers and connecting them in as shown in the figure. This showed the machine to be out of phase with the busbars and the leads were changed back to where they were at first. The cause of the synchroscope giving a wrong indication at first was found to be due to the two transformers to which the instrument was connected being grounded to two different ground wires, which were used as one side of the transformer circuits. This made a difference in potential of about 25 volts and threw the hand of the synchroscope off enough to be the cause of the machine's being connected together considerably out of phase.

With the connections as shown in the figure, the machine to be phased out is started up and brought up to speed. If the connections are correct, all the lamps will go dark and light up together. Should the machine be out of phase, only one lamp will light to full brilliancy at a time.

Another way of phasing out is to connect a three-phase motor to the section of the busbars that the generator being put into service is connected to. The generator can be connected to the busbar and the direction of rotation of the motor noted. After this has been done, the generator can be shut down and the section of the busbar to which the motor is connected can be closed to the other machines on the system and the direction of the motor's rotation again noted. If the motor rotates in the same direction in both cases, the machine that is being put into service is of the right phase relation.

EDWARD SWAN.

Kearney, Neb.

Leaky Discharge Valve on Air Compressor and High Temperatures

In the July 21 issue H. C. Gunson comments on my letter in the May 26 issue, in which I stated that a leaky discharge valve in an air-compressor would not cause an excessive rise of temperature. He is of the opinion that the statement would be qualified on consideration of the entire "leaky" stroke. He asserts that "if the cycle of operations is considered from the beginning of the 'leaky' stroke to the end of the compression stroke, it will be seen that only a portion of the air that leaks through the valve will contribute any cooling effect; that is, that portion which leaks through during the earlier part of the stroke. At three-quarters stroke there will be little travel left in the piston to expand the air that comes in. The slow reversal of motion of the piston allows ample time for a large quantity of air to leak back, and the temperature of this air after its eddies have subsided will be equal to that in the discharge pipe."

I can conceive of some such action, as described in the foregoing, if the piston speed of the compressor were extremely slow, which would preclude any heating at all. But at the speed usual in compressor practice there will be heating due to compression and cooling on expansion. Just what effect the position of the piston

has on expansion is not clear. So long as the piston is free to move, the incoming air will expand and do work, with consequent refrigeration. Were the intake valves closed entirely and the air passed back and forth through the discharge valves (that is, the suction stroke bringing the air in through the leak, and the discharge stroke forcing it out through the opened valve), we would have, in effect, a cold-air refrigeration machine, in which the cylinder on the compression stroke is the compressor and on the suction stroke the expander; and the water-jacket would be the cooler.

The opinion that air can be successively compressed and expanded (in air-compressor practice) until an ignition temperature is reached seems to run afoul of a familiar law of energy. If the work of compression is represented by heating of the air, and a portion of this heat is removed by the jacket water, and by radiation from piping and receiver; and if cooling results from expansion of the air on the suction stroke, how can it be possible to attain any such temperatures as are assigned to this cause, without a creation of energy? Air being compressed is imparted with heat representing the work of compression. If, on expansion and doing work on the piston, the air still retains this heat, subsequent compressions or a few strokes of the compressor will raise the temperature of the air to the melting point of the cast-iron walls of the cylinder.

In an article in the July 21 issue on "Avoiding Compressor Explosions," Thomas O. Organ says: "Suppose air is being compressed to 74 lb. gage pressure and through leaking valves the cylinder should be filled with air at a temperature of 300 deg. before compression, the result will be a final temperature of 833 deg. F., far above the ignition point of the lubricating oil." If Mr. Organ had continued his calculation to include the effect of a number of strokes, instead of one, he might have found his compressor cylinder a molten mass—or found a defect in his hypothesis.

I can agree with Mr. Gunson that any work done on the air must raise its temperature, but I maintain that the accompanying expansion with work on the piston causes a lowering of temperature. There is scarcely a doubt that air compressors are working under nearly all possible degrees of leakiness. If a leak causes high temperature, there should be records of numerous explosions, overheated discharge lines and receivers. I have, in the last three years, overhauled two compressors and which resulted in increased capacity, indicating that both of them had been leaking badly. Neither of these compressors or their receivers showed signs of overheating. The one compressor showed increased capacity by more frequent functioning of the unloader and the other by improved conditions in the quarry which was being worked with air drills.

While discussing air compressors, I wish to relate an experience of a short time ago that might throw light on some failures of compressor equipment to perform as expected. A compressor of 100 cu.ft. capacity was serving a machine shop and, at a distance of 600 ft., a number of packing presses that required 20 cu.ft. of air per min. at 70 lb. pressure. The compressor was set in the machine shop because of the attention that was always available. After the machine was put into service it was found that so long as air was not being used in the machine shop, the presses worked satisfactorily, but when the shop used air, the presses were slowed up and work in both machine shop and pressroom was unsatisfactory. By speeding up the compressor and increasing the pressure to 85 lb. to furnish increased volume, the trouble was corrected. An investigation disclosed that at the time the compressor was purchased, it was assumed that the volume of the air entering the presses would be the same as that leaving the compressor; that is, that the presses would receive the air in its heated and expanded state. The 600 ft. of pipe permitted the air to cool to the temperature of the atmosphere, and which, of course, reduced the volume—the difference being about 50 per cent of the volume at atmospheric temperature.

Los Angeles, Calif. C. O. SANDSTROM.

The several letters that have appeared in recent issues of *Power* on the effect of leaky discharge valves on air compressors, have no doubt proved both instructive and interesting to many readers besides myself.

It is to be hoped that institutions that have the facilities, will prove by practical tests the value of the many theories as to the cause or causes of air-compressor explosions.

The University of California test, mentioned by Frank Richards in his letter in the July 21 issue, proves that two $\frac{1}{16}$-in. holes in a poppet discharge valve had no effect on the temperature of the compressed air.

Increasing the numbers of the holes or increasing their diameter would allow enough air to leak back to fill the cylinder with air at a higher than normal temperature and at a pressure considerably below the discharge pressure. The result of this would be a dangerous discharge temperature. I believe that a leaky piston of a double-acting compressor will have the same effect as a leaky discharge valve. I also believe that a leaky suction valve will allow hot air to return to the suction pipe which may, when the air is drawn from outdoors, have sufficient capacity to retain the heated air returned to it. This heated air in the suction pipe is again drawn into the cylinder and recompressed; this continues until a dangerous temperature is reached.

Insufficient cooling water, a dirty water jacket and an obstructed discharge pipe, are all contributory to high discharge temperature, the latter because of the high pressure.

Some years ago I was sent to inspect an air compressor that had almost entirely wrecked itself. I discovered the cause to be a discharge pipe, so closed with dirt and oil baked hard, that only an opening $\frac{1}{2}$ in. in diameter in a 4-in. pipe remained for the passage of the air.

In two other plants I found that a stop valve had been placed between the relief valve and the air compressor. Regular inspection of the valves and pistons of an air compressor and the discharge pipe will go a long way toward preventing explosions.

Frequent use of the indicator will tend toward keeping the air compressor in its most efficient and safe condition.

The use of soapy water as a lubricant for a few hours every week is common practice and tends to keep the valves and pistons clean. Some air compressors are lubricated with liquid soap only, and do well. A transparent soap jelly thinned with water is used so that it will feed through the ordinary lubricators.

There are readers of *Power* who have had valuable air-compressor experience which I hope they will place at the disposal of their less fortunate brothers.

Long Branch, Calif. E. J. JORDAN.

How Often Should Engineers Change Shifts?

Referring to the discussions in the July 21 issue, on "How Often Should Engineers Change Shifts," the tone of Carl De Garmo's letter led me to believe that he doesn't want his men to rotate for some reason and assumes that they are satisfied. Perhaps they are content, but if a man sees no hope of a change in conditions, he will make the best of things as they are.

Answering Mr. De Garmo's arguments, his contention that it is a good plan to leave watch engineers on the same shift all the time because they get more familiar with the duties and if changed around they get confused and are more likely to become dissatisfied, does not seem reasonable to me.

Surely an engineer should be able to change from one shift to another in the same plant without becoming confused. There may be more work on the day shift than on the others, but each man should be familiar with all apparatus in the plant.

Mr. De Garmo says that it has been his experience that men prefer to stay on the same shift as they do not like having their regular hours broken up. Now my experience has been just the opposite. Men will take the changes in hours in preference to a continuous night shift. I have yet to meet a man who wanted the night shift as a steady thing. There are such men I suppose, but I believe they are few.

As to the irregular habits resulting from changing shifts, affecting the health, the human body is a most adaptable machine and takes care of the changes easily. Put this argument down as a mental condition. Some men often speak of the changes in living necessitated by changing shifts, but just tell the man who complains, that he can quit changing shifts if he will work the night shift all the time and see how quickly he will refuse, especially in the summer.

As for the change giving the chief opportunity to observe the men, there is nothing that can take the place of personal observation in enabling a chief to judge his men fairly. The idea of calling a man "to the office" is a bit too formal for the average plant. Instructions can be given without calling a man "to the office."

I agree with Mr. Spencer that the chief should canvass the men and then adopt a schedule that will satisfy the majority.

The usual change hours are at 8 a.m., 4 p.m. and 12 p.m. or at 7 a.m., 3 p.m. and 11 p.m. Of these two schedules I prefer the former, for various reasons. It is nice for the day man to go home at 3 p.m., but it cuts the afternoon too short for the man coming on at 3 p.m. I would much rather have a night shift begin at 12 than at 11 p.m. because it gives the night man ample time to go to a theater or social affairs. Also, it gives more time to the night man who takes a nap before going to work. In some plants the shifts change at 6 a.m., 2 p.m. and 10 p.m., but this schedule is neither common nor desirable.

For a change schedule, I believe that the following is a better arrangement than that proposed by Mr. Spencer:

	7 A.M.	3 P.M.	11 P.M.	7 A.M
Up to Saturday		A	B	C
Sunday on		B	C	A

By this method the day man A gets the long change, being off from 3 p.m. Saturday until 11 p.m. Sunday. In short, the day shift is the best of all, so is it fair to give one man the best of it all the time instead of dividing the good and bad impartially?

Dawson, N. M. A. L. CONWELL.

Fineness of Pulverized Coal

On page 140 of the July 28 issue there is an extract from the Bureau of Mines Bulletin No. 237, which reads in part: "In the past it was thought that to insure efficient combustion with powdered coal it must be pulverized so that 85 per cent passed through a 200-mesh and 90 per cent through a 100-mesh screen. The tests at Lakeside and subsequent experiments with coarsely pulverized coal indicate that with this type of furnace and burners, it makes little difference what percentage of the coal passes through 100- to 200-mesh screens if nearly all of it passes through 50-mesh."

There is nothing particularly new about 85 per cent through 200-mesh being unnecessarily fine for good combustion results. The writer called attention in the Aug. 23, 1921, issue of *Power* to the results of experiments run early in 1920 on moisture content and fineness, which showed that for peat, 50-60 per cent through 200-mesh was ample fineness; 65-70 per cent for lignite; and 73-80 per cent for bituminous coal. The results indicated that the fineness necessary varied directly with the specific gravity of the fuel. With light fuels, such as peat, quite large particles could easily be carried along in the air currents until they were ignited; while with a heavy fuel, such as anthracite, the particles had to be considerably finer to prevent their dropping out unburned.

It is, however, significant that the Bureau of Mines has found that the coarseness can be carried down to 50-mesh product, and if experience bears this out, the fact should have an important bearing on the use of pulverized coal. One of the main stumbling blocks in the past has usually been preparation costs, and the cost of pulverizing is a major item—probably the largest single item. It costs more to crush coal from 160-mesh to 200-mesh than it does to take it from coarse down to 160-mesh, and if the fan drawing the coal from the pulverizer can be set to lift out 50-mesh product without letting more than a nominal portion become ground finer, then there will certainly be a tremendous reduction in the costs of pulverizing.

At the Spring meeting of the American Society of Mechanical Engineers, L. V. Andrews presented a paper in which he showed that the percentage through a 200-mesh screen did not indicate a great deal about the actual fineness of powdered coal. He pointed out that a sample containing 85 per cent by weight of coal which would pass through a 1,000-mesh screen might show the same analysis with 100- and 200-mesh screens, that a sample containing 85 per cent of a product, which would pass 200-mesh but would be held on 250-mesh, would show. He urged a microscopic study of pulverized coal samples to show the distribution of sizes finer than 200-mesh, in order to determine the point at which to lift coal out of the pulverizer without allowing it to be ground too finely.

If 50-mesh product is to become the new standard, such precautions would be entirely unnecessary, and a coarse grinding unit would be sufficient. But possibly the Bureau did not intend that its findings should be interpreted too literally. W. J. RISLEY, JR.

North Glenside, Pa.

Questions and Answers

Conducted by

Franklin Van Winkle

Effect of Adding Inside Lap to D Slide Valve

What is the effect of inside lap of a D slide valve?
J. J. S.

The effect of adding either inside or outside lap is to cause the port to be opened later and close earlier. Hence without making any other changes in the valve or its setting, the effect of adding inside lap to a D slide valve with outside admission, would be to open the port with release of the cylinder steam at a later point of the stroke and close the port from the exhaust with earlier compression.

Pump Discharge Greater than Piston Displacement

Can the amount of water delivered by a pump exceed the piston displacement? *E. R. L.*

This is likely to occur with slow-running pumps and large and long suction pipes. When during a suction stroke the velocity of the pump piston is reduced, there may not be corresponding reduction of the velocity of the suction water and the inertia of the suction water may be so great as to create sufficient pressure for some of the water to be delivered from the suction side of the piston through the discharge valves. The quantity thus discharged may be in excess of the piston displacement.

Figuring Brine Cooling Surface

What is considered the average amount of heat transfer per square foot of surface per hour per degree of temperature difference between ammonia and brine in figuring the ice-making capacity of shell and brine coolers? Is it customary to include in the computation the surfaces of the shell and heads of the coolers on an equal basis of heat transfer as the surfaces of the tubes?
E. H. T.

A fair heat transfer is 180 to 200 B.t.u. per sq.ft. per deg. F. temperature difference per hour. Heads will be disregarded in figuring surface, inasmuch as no cooler should be designed so closely that the heads must be included in determining the cooling surface.

Economy of Returning Condensate of Heating System to Boiler

Our heating system is supplied with live steam. The condensate at present discharged as waste amounts to two gallons in three minutes and has a temperature of 206 deg. F. If the condensate could be returned to the boiler, what would be the saving in pounds of coal burned per hour? *J. F. B.*

The saving would depend on the present temperature of boiler-feed water, the pressure of steam generated in the boiler and the number of pounds of boiler feed water at present evaporated per pound of coal. The saving would apply only to that proportion of boiler output whose weight is the same as that of the water returned from the heating system.

Ordinarily, the saving from increase of temperature of the feed water amounts to about 1 per cent for each 10 deg. F. increase of the boiler feed-water temperature. On this basis, if the present feed-water temperature is 50 deg. F. and the returns could be delivered to the boiler at the temperature of 206 deg. F., the increase of temperature would be 156 deg. F. and the fuel saving would be about 15.6 per cent for the evaporation of the same quantity of feed water.

Returns amounting to two gallons in three minutes, or 40 gallons an hour, would weigh $40 \times 8.33 = 333.3$ lb., and if the boiler evaporated 7 lb. of water per pound of coal, the present coal required to generate 333.3 lb. of steam would be $333.3 \div 7 = 47.6$ lb., and the coal saving would amount to 15.6 per cent of $47.6 = 7.4$ lb. of coal per hour, which of course does not include any additional cost or charges for returning the condensate to the boiler.

Fuel Cost per 1,000 Lb. of Steam

With coal costing $7.70 per ton of 2,000 lb. and evaporation by the boiler of 7 lb. of water per pound of coal, what is the fuel cost for the generation of 1,000 lb. of steam? *J. J. S.*

One ton of the coal would evaporate $2{,}000 \times 7$ lb. of the feed water and therefore the fuel cost per 1,000 lb. of steam would be $\$7.70 \div (2{,}000 \times 7) \times 1{,}000 = \$0.55$. As indicated in the statement, when the price of coal is quoted per ton of 2,000 lb., the fuel cost per 1,000 lb. of steam is readily found by dividing the cost of coal per ton by two times the number of pounds of feed water evaporated by a pound of coal.

Operation of Overloaded Corliss Engine

We have a single-eccentric Corliss engine used for driving a flour mill on one side and a wood-working shop on the other. Generally the engine runs nicely, but occasionally the governor rises and falls and the valve gear "skips" with alternation of slowing down and racing of the engine. What is the cause and remedy? *F. A. M.*

The irregular action undoubtedly is due to temporary overloading of the engine, or throttling, or drop of the boiler pressure requiring later cutoff than is within control of the governor. With a Corliss valve gear cutoff with disengagement of the valve must occur before the wristplate has swung farthest to one side of the central position and, with a single-eccentric Corliss engine, in which some angular advance is given to the eccentric to obtain compression by closure of the exhaust valves before the end of the stroke, the eccentric will be on the line of dead centers with the limit of

travel of the wristplate, and the valves will have their greatest displacement when the crank has moved through less than 90 deg. and before the piston is at half stroke. If the detachment gear has not been released before the valve has reached its greatest displacement, the valve will not be disengaged but will remain under the control of the linkage connecting it to the wristplate and cutoff will not occur until near the end of the stroke. Then the time the valve will be closed will depend on the lap of the valve and angular advance of the eccentric, as with a plain slide valve.

To obtain smooth running, the load must be kept within the range of the governor action, and it may be possible to meet the present overload by supplying the engine with steam of higher boiler pressure.

Combined Operation of Deep-Well and Belt-Driven Plunger Pump

Will a belt-driven plunger pump work satisfactorily if it receives water under pressure directly from the discharge line of a deep-well pump? *R. E. B.*

The pumps cannot operate satisfactorily unless there is some kind of an expansion chamber or relief valve between them, as both pumps cannot be regulated to maintain the same rate of discharge and, without compensation, either or both pumps will operate most of the time with slippage and pounding. The best method of connection would be to have the first pump deliver to a tank or standpipe from which the belt-driven pump can take its supply.

Purging Ammonia System

For purging our ammonia system of air, we have a hose connected to the top of an oil trap and into a barrel containing about 30 gal. of water. When purging, the water rumbles as when steam is discharged from a hose into a barrel of water. There is little noise or odor until a fog appears, and then there is a cracking noise. We just crack the purge valve. Is this method correct? *W. F. C.*

Usually, the top of the oil trap is not the proper place for the purge connection. Purging should be done at the condenser. The air will cause the water to circulate as you describe. The crackling sound is caused by the sudden absorption of the ammonia vapor by the water. It is preferable to purge at the top of the condenser after shutting down and cooling the condenser as much as possible. This will condense most of the ammonia vapor in the condenser, and the ammonia lost with the purged air is less than when the plant is purged while running.

Duplex Feed Connections of Little Advantage to Circulation of H. R. T. Boiler

Would the circulation or efficiency of a horizontal return-tubular boiler be increased from feeding by a pump through the front head of the boiler, and returning high-pressure condensate into the blow-off with a return trap; or would the same result be obtained by discharging the trap into the feed line from the pump? *H. W. B.*

Undoubtedly, there would be a small increase of circulation and improvement in efficiency from introducing feed water at different parts of the boiler. However, in ordinary operation of a return-tubular boiler, the automatic circulation of the boiler water is so much greater than the increase of circulation that would be introducing the feed water at different points that the advantage probably could not be discovered by the most accurately conducted boiler tests. Hence, for all practical purposes the same results would be obtained by discharging the trap into the feed line from the pump.

Relative Position of Pulleys for Quarter-Turn Belt

How should a machine pulley be placed to receive power transmitted from an overhead lineshaft by a quarter-turn belt? *R. W.*

The position for the machine pulley, with respect to the pulley on the shaft, depends on the relative direction of rotation of the pulleys. In any event the machine pulley must be in such a position with respect to the overhead pulley that, when a plumb line is hung over the falling side of the overhead pulley at the middle of its face, the line must touch the rising side of the machine pulley at the middle of its face. Referring to Figs. 1 and 2, in which the same letters of reference are used, suppose *S*, as shown in Plan 1 or Plan 2, represents

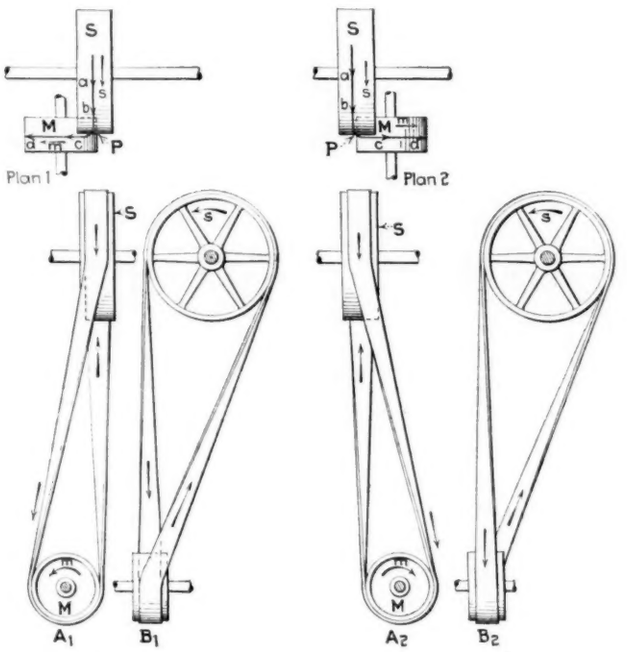


Fig. 1 *Fig. 2*

Placing pulleys for quarter-turn belt drive

the overhead pulley whose direction of rotation is indicated by the arrow *s*, and it is desired to drive the machine pulley *M* in the direction indicated by the arrow *m*. Then the middle of the face of the rising side of pulley *M* must be vertically under the middle of the face of the falling side of pulley *S*, that is, a plumb line hung over *S* at *P*, Plan 1 or Plan 2, should touch the rising side of pulley *M* at the middle of its face. It should be noted that where *s* and *m* indicate the directions of rotation, whether per Plan 1 or Plan 2, the belt must pass over the top of the overhead pulley and thence to the far side of the machine pulley as indicated, in each plan, by the arrows *abcd*. The arrangement and direction of the belt for the relative directions of rotation of the pulleys per Plan 1 are shown in elevation at A_1 and B_1 and for rotations per Plan 2 the elevations are shown at A_2 and B_2.

New and Improved Equipment

The "Unipulvo" Unit Pulverizer

A recent addition to the growing family of unit pulverizers has been developed by the Strong-Scott Manufacturing Co., Minneapolis, Minn. In common with most unit pulverizers the new machine is a self-contained unit designed to serve the individual boiler.

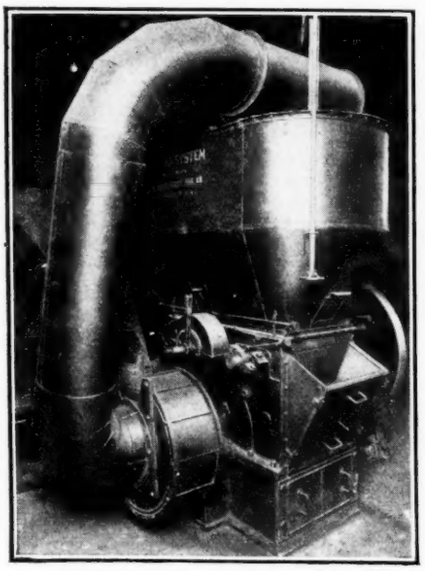


Fig. 1—New Unipulvo unit pulverizer installation

Fig. 1 shows an installation in Minneapolis including the pulverizer, fan and the horizontal, flared burner leading into the boiler furnace. The general design is more clearly seen in the sectional elevation, Fig. 2.

From the large coal hopper at the top the coal passes through a crushing and mixing roll designed to reduce the over sizes found in all screenings and to mix the fine with the coarse so as to present uniform material to the feeder roll which introduces the coal into the pulverizer. The crushing element will handle 3-in. lumps. It operates against hinged breaker plates or gates held down in working position by counterweights so that in the event any uncrushable material comes down with the coal, the plate yields, allowing the object to pass, or if it is a large object, ejecting it from the machine. This provision eliminates the need for magnetic separation.

The feeder roll, shown a little to the left of the crusher, is of the wing type actuated by a pawl-and-ratchet mechanism that can be set to feed at any desired rate. This mechanism is inclosed in a dustproof case and runs in oil. To protect it from overstrain from the passing of large material, the feeder roll operates against spring-retained screed plates. From the feeder the coal passes down in front of the rotor, which is of the swing hammer type carried on ball bearings. It revolves in an enlarged chamber having a bed of coal 6 to 8 in. deep at the bottom, a clearance of 4 to 5 in. at the back and 3 in. at the top. At the front or the air-inlet side the chamber is entirely open.

All the air required for combustion is taken into the rotor chamber with the coal feed and is swept around the rotor once to discharge at the top into the expansion and separating chamber. It will be seen that the rotor acts as a closed-end fan and the hammers as fan blades to create an intense cyclonic air action in the rotor chamber. The sweep of the air draws the oversize returning from the separating chamber, as well as the coal fed into the mill, around the rotor and against the stationary coal.

The air-swept coal mixture passes around the rotor once, being discharged in a continuous stream into the separating chamber, from which it passes through tangential vanes into a cone in which the vortex action set up separates the fine from the oversize. The fines go out with the air to the exhaust fan, thence to the burner. The coarse particles drop back for further grinding. The amount of air taken through the mill is regulated by a blast gate in the fan intake, and the fineness of pulverizing is controlled by elevating and depressing the cone in the separating chamber. Tramp metal or rock from the coal accumulates in the bottom of the machine out of contact with the rotor, where it can be removed from time to time even during operation.

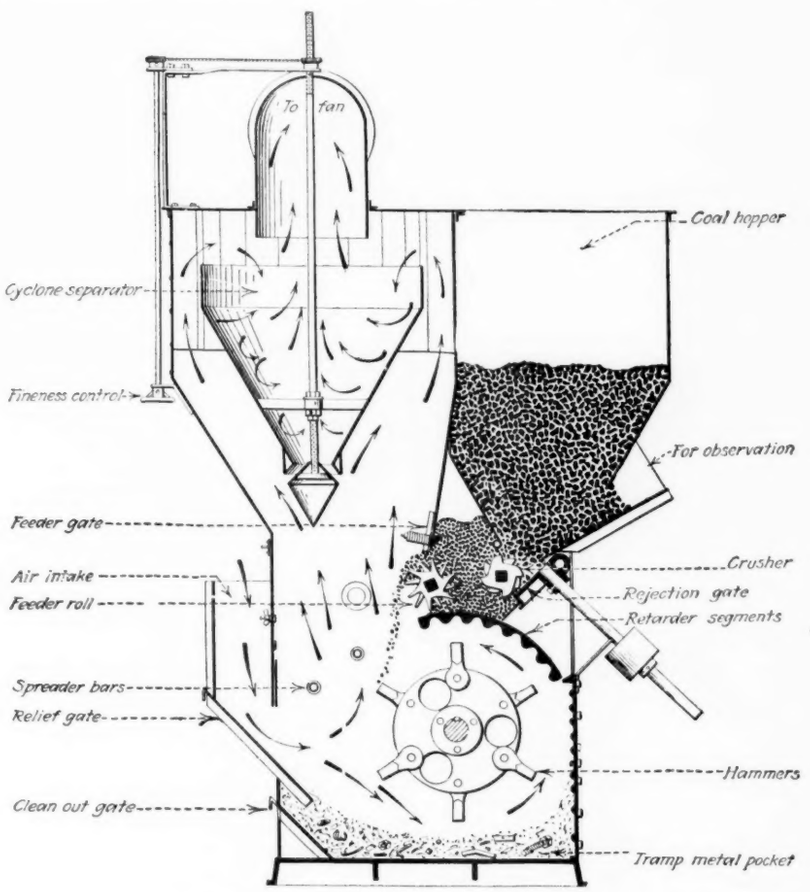


Fig. 2—Sectional elevation showing working principles of pulverizer

In the design of the machine the crusher and feeder elements have been made unit castings which are slipped

over square shafts and replaced by pulling the shaft out through the side of the casing. By pulling two slides, the entire rotor chamber is made accessible. To facilitate hammer replacement, the pins on which they swing come out through the side of the casing. From the main shaft to the crusher and feeder shafts, a friction reduction drive is used.

According to H. G. Lykken, the inventor, the Unipulvo mill has been covered by extensive patent applications in this and other industrial countries.

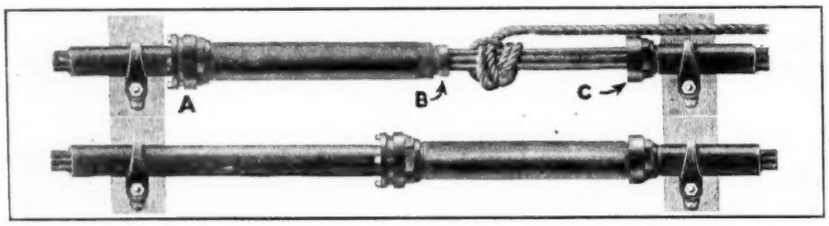
Conduit Pull-and-Splice Fitting

Where wires are pulled into long lengths of conduit, it is usual practice to put in pull boxes at intermediate points, so that the wires may be pulled from more than one point and the strain on the wire will not be sufficient to injure the insulation.

To reduce the cost and the space required for pull boxes, the V. V. Fittings Co., Philadelphia, Pa., has developed and placed on the market what is known as a "Straight Line Pull and Splice Fitting," shown in the figure. This fitting consists of three parts: A stuffing box and gland with packing *A*, a bushing *B*, and an end connector *C*. For the connection between the stuffing box and end connector a length of conduit is used, having a diameter large enough to go over the conduit that the wires are pulled into. For example, if the main conduit is 1¼ in. in diameter, a piece of 1½ in. will be used for the sliding sleeve between the fittings.

The space for pulling the wires can be made almost any length up to 9 ft. which would require a standard 10-ft. length of conduit for the sliding sleeve. In applying this fitting, the stuffing box and gland *A* are screwed onto the end of the sleeve and they are pushed on over the conduit out of the way. Then the bushing *B* is put on the end of the main conduit in front of the sleeve and the end connector *C* is placed on the end of the conduit on the opposite side of the pull opening.

After the wires have been pulled in, the sleeve is moved up over them and screwed into the end connector and the stuffing-box gland tightened so as to make a water-tight connection, as shown in the lower figure. In addition to using the fitting as a pull box, it may also be used as a junction box in which the wires are spliced. The use of this fitting not only provides greater convenience in pulling in the wire than do pull boxes, but it also occupies much less space. The fittings are made of cast iron, black or galvanized finished in standard conduit sizes from 1 to 6 inches.



Fitting replaces pull boxes in long conduit lines where cables have to be pulled from more than one point

The "CombustiKator"

The arrangement of furnace shown in the accompanying illustrations has been developed by the CombustiKator Co., a division of the CoKal Stoker Corp., of Chicago. The design was first applied to low-pressure heating boilers, but later has been used to advantage on high-pressure return-tubular, firebox and even horizontally baffled water-tube boilers.

The "CombustiKator" consists of an inside refractory hopper of special design, standard inclined CoKal grate and the dumping mechanism. Everything is contained within the boiler except the two levers used respectively for feeding coke to the fuel bed and for dumping ashes. The outstanding feature of the "CombustiKator" is the refractory arch dividing the furnace into a coking chamber and the furnace proper. It is, in effect, both a gas producer and a stoker, where the gas is produced and burned over the incandescent coke on the grate. Because of the gas-producer principle employed, bituminous coals are said to be burned smokelessly.

In operation, coal is thrown through the firing door against the wall of the coking chamber, which is also the refractory arch. Here the coal is coked, and the volatile or hydrocarbons evolved escape through passages in the arch, where they are preheated and in being discharged over the incandescent fuel bed are burned to complete combustion. The coal does not reach the grate or active combustion zone until it has been thoroughly coked. Combustion is smokeless and the efficiency high.

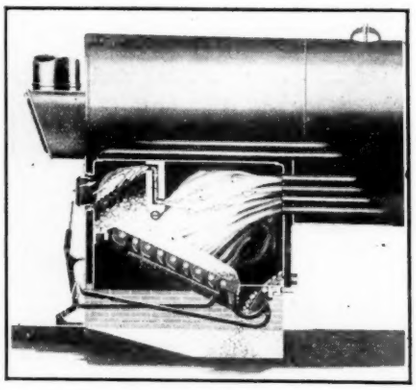


Fig. 2—CombustiKator serving return-tubular boiler

Cannot Disturb the Fuel Bed

The reasons advanced for the results obtained are: That the fireman cannot interfere with the fuel bed; that the coke bank keeps the passage to the furnace closed, so that cold air does not rush into the furnace every time the firing door is opened; that no hand cleaning of the fire is required as the intermeshing grate bars keep the fuel bed opened up and free from clinker, and that the volatile, after being preheated, is burned completely without smoke.

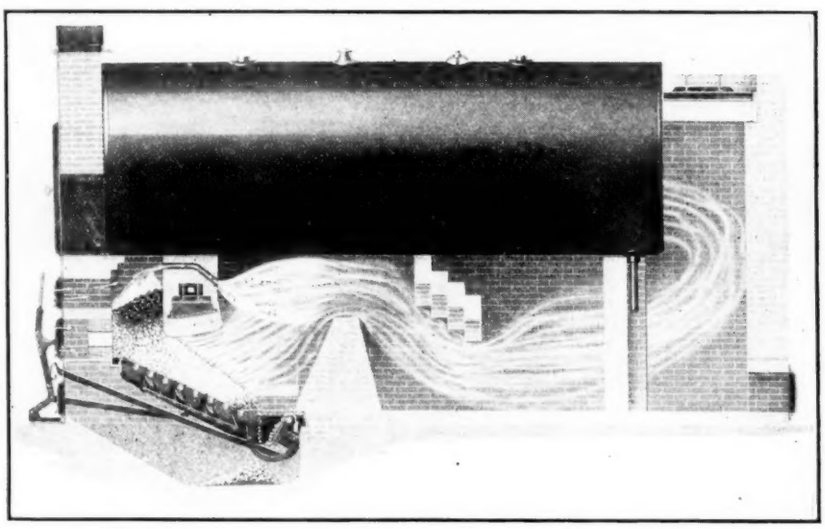


Fig. 1—CombustiKator under firebox boiler

Auxiliaries and Auxiliary Drives for Steam-Electric Generating Stations*

By A. L. Penniman, Jr., and F. W. Quarles†

AUXILIARY drive should so influence station design that the largest amount of net electrical energy may be generated for the least expense in money, including fixed as well as operating charges, consistent with reliability.

Classification of Auxiliaries

Auxiliaries fall into three general classes, with respect to the necessity for an unfailing power supply:

a. Equipment little affected by an interruption.

b. Auxiliaries that are required only when the station is carrying load, and must be shut down upon loss of load.

c. Auxiliaries that are intimately related to the production of power, whose stoppage may result in disaster or prolonged interruption.

Methods of Driving Auxiliaries

The steam-turbine drive is well adapted to this service because of its reliability. Non-condensing turbines are generally used. The station heat balance is difficult to maintain with variable load, but may be improved by extracting steam from the main turbine unit. Heat loss in transmission of steam to small turbines has always been of consequence and the advent of higher steam pressures and temperature increases this loss, and also affects the reliability of small turbines, and increases maintenance and first cost.

Motors with power obtained from the main station bus, have the advantage of simplicity of operation and installation. A serious disadvantage is the effect of an interruption to the main station bus shutting down the plant. Some designers have met this condition by the use of duplicate equipment, steam and motor driven, or dual drive.

The house turbo-generator was a logical step in providing an unfailing supply of power for motors driving auxiliaries. It has the better economy of a larger turbine. On the other hand, the efficiency of the house turbine is not as high as that of the main unit.

With any system of motor drive, speed regulation is readily effected without resort to gears. The direct-current house turbo-generator sacrifices this advantage because direct-current generators of the size commonly needed require a reduction gear between the house turbine and the direct-current generator.

The house-turbine unit has the problem of controlling its output to suit the economical requirements of feed-water heating without running in parallel with the main station bus, unless through a loosely coupled connection. One method has been to use transfer motor generators. Another is the use of two power buses, one fed from the main station bus, the other from the house turbo-generator. Load is transferred by switching individual motors or groups of motors from one source of supply to the other.

Direct-Connected Auxiliary Generator

The auxiliaries serving large reciprocating engines were sometimes driven from the shaft of the engine. This is paralleled today by an auxiliary generator connected to the shaft of the main generating unit. Power obtained in this manner is highly dependable and is affected to a negligible extent by an interruption on the main station bus. This has the advantages of the house-turbine unit without the turbine part of the unit, and the prime mover is a turbine more efficient than the smaller unit. When this is combined with the use of the regenerative cycle, heating the feed water by bleeding steam from the main unit, it assumes a favorable aspect. In effect, we obtain byproduct power at small cost. The farther the steam is allowed to travel through the main turbine before extraction, the greater is the by-product power.

If the direct-connected auxiliary generator is a direct-current machine, simply an enlarged exciter, the advantages of direct-current motors are obtained for those services that are benefited by speed variations, without loss for energy conversion. However, there are difficulties in the design of such high-speed direct-current generators, and manufacturers have been loath to furnish them. Consideration should be given to the shaft-driven alternating-current generator. The design does not require special treatment, and it can be designed for high efficiency, low cost and low maintenance. Its action under short-circuit is more favorable. Safeguards against disturbances are more readily applied.

Since the power supplied from the shaft-driven alternating-current generator is dependable, it may be used to drive a motor-generator to furnish direct current for excitation and for power supply to such motors as may be deemed desirable. The direct-current generator in this case need not be special. Unless sufficient gain results from the use of direct-current motors to offset the conversion losses, it is advisable to use alternating-current motors, particularly of the squirrel-cage or wound-rotor type, since they are cheaper both in first cost and in maintenance and are inherently reliable.

Another method for securing power for the essential auxiliaries which has gained considerable favor, is the spinning house turbo-alternator, which is normally driven electrically with no steam passing through the turbine, but which will automatically pick up the essential load in event of failure of the main station voltage. This arrangement calls for additional capital investment in turbines, piping and transformers, while there is a definite operating loss.

Still another scheme is to connect the auxiliary power transformer to the main generator leads and insert a suitable reactance between t' is point and the main bus. This has the advantage of low first cost, but until it has been given the test of protracted service, does not seem to be as reliable nor nearly as desirable from an operating point of view as the direct-connected house generator.

Motors

All motors for power-plant use should be acid- and moisture-proof, and all turbine auxiliary motors should be provided with splashproof housings. Motors installed on the boiler-room floor should be totally inclosed, preferably being fitted with navy-type frames and end bells.

When motors of adequate size with suitable insulation are used, the question of motor outage reduces itself to the question of bearing failures. It has been the writers' experience that the failure of motor bearings can be practically eliminated by the use of self-aligning ball and roller bearings. It is essential that proper provision be made for excluding moisture and dirt from the bearings as well as for keeping the lubricating medium within the bearing housing. The bearing mounting of the cartridge type, when equipped with labyrinth sealing rings, provides a water-, oil-, and dust-tight mounting, which will be entirely satisfactory when fitted with bearings having an ample factor of safety. The factor of safety should be not less than 12 and preferably higher on the basis of catalog ratings. Experience is cited to show the great value of the use of self-aligning ball and roller bearings for motors and pumps. Such bearings have been used on pumps up to 45,000 gal. per min. capacity with complete success. A new station is just going into service, equipped with self-aligning ball or roller bearings on all motors over ½ hp. in size, while all the condenser pumps are fitted with roller bearings.

*Abstract of paper presented at the twentieth annual convention of the Association of Iron & Steel Electrical Engineers, Philadelphia, Sept. 14-19, 1925.

†The authors are respectively superintendent of steam stations and assistant superintendent of steam stations, of the Consolidated Gas, Electric Light & Power Co., Baltimore, Md.

Industrial Engineers Elect Officers

The Society of Industrial Engineers, 606 South Dearborn St., Chicago, Ill., at its annual meeting on Sept. 29 held in Detroit, elected the following national officers: President, Walter R. Rittman, Carnegie Institute of Technology, Pittsburgh; vice-president in charge of research, Perry A. Fellows, city engineer, Detroit; vice-president in charge of education, Charles B. Gordy, University of Michigan, Ann Arbor; vice-president in charge of promotion, W. W. Nichols, D. P. Brown & Co., Detroit; treasurer, S. F. Mitzner, H. N. Stronck & Co., Chicago; secretary, George C. Dent, 608 South Dearborn St., Chicago. The thirteenth national convention of the society will be held at Philadelphia, Pa., Jan. 16-18, 1926.

Improvement in Designs of Synchronous Motors*

SYNCHRONOUS machines employed as power motors were little thought of 20 years ago. The ever-increasing inductive load on power systems brought about power-factor problems. As the synchronous machine was fitted for the purpose of improving low lagging power factor, it came into use as a synchronous condenser, which is an over-

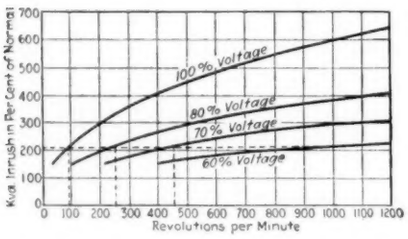


Fig. 1—Relation between inrush kva. and speed of synchronous motors

excited synchronous motor floating on the line. This motor was also used in connection with motor-generator sets when the starting duty was light. It soon became evident, however, that the synchronous motor had inherent characteristics for a power motor and especially so when the speed was low. It is practically out of the question to build efficient slow-speed induction motors, and besides the price is almost prohibitive.

What put the synchronous motor in the forefront was the fact that, besides being an efficient and rugged power motor, it could at the same time correct the bad effects of low power fac-

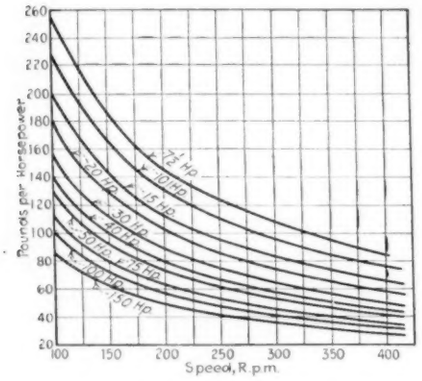


Fig. 2—Relation between weight and speed of synchronous motors

tor. Fifteen to twenty years ago the only serious objection against the synchronous motor was poor starting characteristics. Treating the synchronous motor as a motor and getting away from modified alternator designs has improved its starting characteristics to such an extent that, at the present time, synchronous motors of standard manufacture compare very well with the ordinary squirrel-cage motors. Synchronous motors of special design have better starting characteristics even than slip-ring type induction motors.

A large field of application was found, several years ago, for synchronous motors by direct connecting to slow-speed reciprocating air and ammonia compressors. Provided with full-size bypass a compressor requires but light starting and pull-in torque. In most cases the necessary flywheel effect can be incorporated in the synchronous motor's rotor and makes a compact and efficient unit.

Kilovolt-Ampere Inrush to Synchronous Motors

Fig. 1 shows the kilovolt-ampere inrush in per cent of normal, at various applied voltages for different speeds of 60-cycle synchronous motors. These curves are based on normal motor ca-

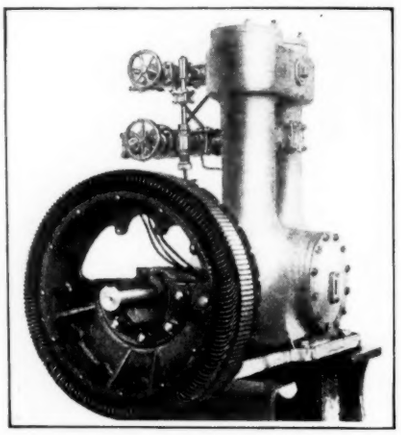


Fig. 3—Stator of 30-hp. motor in place on compressor

pacity of 300 kva. and terminal voltage not to exceed 2,200 volts. For larger capacities the kva. inrush values will be slightly lower and for smaller motor capacities they will be somewhat higher. Note that a 90-r.p.m. synchronous motor thrown direct on the line will draw no more current than a 250-r.p.m. motor started on the 80 per cent tap, than a 450-r.p.m. on the 70 per cent tap, or a 900-r.p.m. on the 60 per cent tap, as indicated by the dotted lines.

Investigation of such inrush characteristics at starting led to the "across the line" synchronous motor with full automatic control. Lately, this method of starting has become almost standard for slow-speed motors, as it eliminates autotransformers and tandem switches. It also eliminates the human element, as the operator had to use his best judgment as to when to throw the motor over on full voltage. It also insures against mechanical shocks which may wreck a direct-connected unit and against line disturbances which occur when a synchronous motor is disconnected and reconnected to the line during the starting period. Synchronous motors have been successfully applied for driving:

Motor-generator sets	Plunger pumps
Frequency converters	Screw pumps
Air compressors	Blowers and fans
Ammonia compressors	Tube mills
Pulp grinders	Flour mills
Jordans	Rubber mills
Stone crushers	Cement mills
Centrifugal pumps	Steel and copper rolls
Lineshafting	For operating as synchronous condensers
Conveyors	

Modern design has brought out lines of special synchronous motors. The writer recently designed a line of syn-

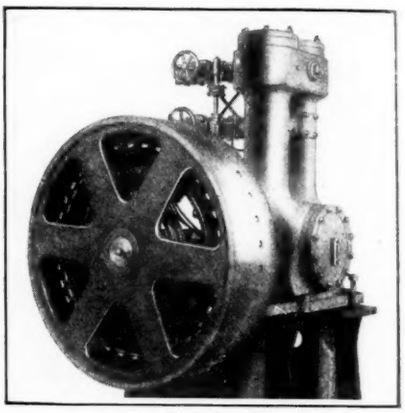


Fig. 4—Flywheel-type 30-hp. motor completely assembled on vertical compressor

chronous machines known as the "Flywheel Type." These machines are adapted equally well for alternators and synchronous motors for direct connection to small reciprocating engines forming self-contained units, and patents have been applied for.

In the field of refrigeration and compressed air there is an increased tendency to employ synchronous motors direct connected to small compressors. Among the advantages of the direct-connected slow-speed synchronous unit over the belted high-speed induction motor are: Higher efficiency, less space requirements, lower upkeep and preferential power rates.

First Cost the Main Obstacle

The main obstacle, however, in making the small direct-connected synchronous-machine installation standard practice has been the first cost of the motor or alternator. The reason for this becomes apparent when the weight characteristics of a line of smaller 60-cycle engine-type synchronous motors of standard design are considered. Referring to Fig. 2, pounds per horsepower are plotted against speeds for various horsepower ratings. It will be noted that the weight in pounds per horsepower increases rapidly as the speed decreases.

Direct-connected engine-type synchronous motors for small compressors, both of the vertical and horizontal type, have been built as "overhung" motors. To make these new flywheel-type machines as compact and inexpensive as possible and also to overcome other difficulties experienced with the old type of construction, these new synchronous machines are designed with the rotating field outside the stator. The stator

*Abstract of a paper, "The Evolution of the Synchronous Machine from the Early Days to the Present," by Theo. Schou, chief engineer, Ideal Electric & Manufacturing Co., presented at the 43rd Annual Convention of the National Association of Stationary Engineers, held in St. Paul, Minn., Aug. 31 to Sept. 4, 1925.

is mounted in a cradle support forming a part of the compressor or engine casting, and the female fit of the cradle seat is machined to gage at the same setting as the bearing housing is machined. To produce a true bore and also to prevent any possible shifting of the stator in an upward direction, the cradle seat is extended somewhat above the half-circle, thus insuring a locked condition of the stator when it is bolted to the cradle support. As the male fit of the stator spider is machined to exact cradle fit, a uniform air gap is obtained, and there is no possibility of any relative movement of the stator or rotor. This is an essential feature

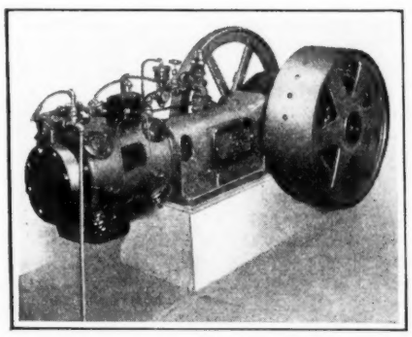


Fig. 5—Flywheel-type 100-hp. motor assembled on horizontal compressor

with small air-gap synchronous machines.

Fig. 3 shows the motor's stator in place on a small compressor.

The rotor which revolves outside the stator has a pulley fit on the shaft extension, and is locked in place by a key. As all vital parts are inside and protected, this rotor construction does not necessitate such careful handling as the old type. The rotor may roll on the floor without in any way damaging the poles or squirrel-cage winding. The complete assembly of a 30-hp. 200-r.p.m. 60-cycle motor direct connected to a vertical compressor is shown in Fig. 4.

A flywheel-type 100-hp. 257 r.p.m. synchronous motor, driving a compressor is shown in Fig. 5. It should be noted that although an auxiliary wheel is shown on the horizontal compressor, this is not needed for flywheel purposes, as the normal flywheel effect of the rotor is 50 per cent in excess of that required.

This new construction eliminates many of the difficulties encountered with the old standard type. There is more space between the poles, the field leakage is decreased, and moderate exciter capacity is the result.

To obtain proper flywheel effect has become an easy matter. With approximately the same rotor weight as the old standard, twice the flywheel effect is obtained, because of the greater rotor diameter. Should the normal flywheel effect be insufficient, which rarely is the case with this type of construction, more rim section is added, and thus the flywheel-effect problem is reduced to bearing limitations of an overhung machine.

In general, this motor construction looks much like an ordinary pulley, its face is crowned, and an exciter or any other auxiliary machine may be belted to the rotor. The unit is self-contained, the motor forming an integral part of the compressor. This facilitates shipping and freight conditions; less parts, less cratings and less weight. It is shipped to the customer ready to run; a motor foundation is not required, and no motor erecting work is necessary.

Limitations of Superpower*

By G. C. NEFF†

SO MUCH has been said about superpower and so much has been promised by popular articles in the lay press that most people look upon it as some new discovery that has magic powers and is about to revolutionize industry in this country. In discussing the subject we are prone to theorize and dream about some mysterious inexhaustible supply of energy that costs little or nothing, without attempting to learn the facts regarding the situation.

In practically all such discussions water power plays a very important part. The mere fact that a water-power site exists and there is located at the site a given number of theoretical potential horsepower does not justify the conclusion that a water-power development at that location can be developed economically, for the economic development depends upon many factors.

Electrical energy is now being produced in many steam plants at less cost per kilowatt-hour than in many hydro-electric stations. Only comparatively few of the hydro-electric stations of this country can successfully compete in low cost of generation with the large, modern, well-located steam plants of today. This statement may seem absurd, since it is generally believed that water power always generates electricity cheaper than steam plants for the reason that steam plants burn coal and water-power plants do not. This reasoning is not sound, because it does not take into consideration all the items of cost, but only one, namely that of fuel cost. There are other items of cost which, in many cases, more than offset the fuel cost, and therefore must be given consideration.

Very often in listening to the average discussion of the wonders of superpower we are told that through the development of huge water powers and steam plants connected to transmission lines, the farmers of this country are going to receive energy at very low cost.

Electricity cannot be transmitted from a generating station to the market without some considerable loss. When electricity is carried over transmission lines, some of it is lost and this loss must be made up by the generating station, which calls for additional capacity and for additional coal. The transmission line costs money, and there are certain fixed charges on this line which have to be met, and this also adds to the cost of electricity. The lines in time will need to be replaced and this adds to the cost. These items of cost of transmission often much more than offset the advantages that can be gained in locating the steam-generating plant at the mine, even if there is sufficient water available.

In a discussion of the paper, S. B. Hood, Northern States Power Co., stated that he was particularly pleased to note the warning sounded as to the limitations of superpower and where it may lead to if we see fit to go off into day dreams instead of coming down to brass tacks on every-day economics.

Suppose the Riverside steam plant be moved from Minneapolis to Duluth, thereby saving around $232,000 per annum in coal freight. Now certainly a steam plant at the Duluth docks will have ample water for condensation or makeup requirements and therefore nullifies one of the arguments against mine-mouth plants. To build a double-circuit transmission line between Duluth and Minneapolis would cost in round figures $2,250,000. This would have a transmission loss of around 10 per cent, so the steam plant would have to be that much larger to fill requirements. This means another $750,000 investment, or a total of $3,000,000.

On this investment 12 per cent per annum must be earned to cover fixed charges alone, or $360,000 per annum. On top of this the cost of generating this 10 per cent transmitting loss would be another $30,000. Here we have, then, $390,000 to offset freight savings of $232,000. In other words, we go in the hole $158,000 per annum without counting the operating and maintenance cost of our transmission line. This would be an investment of around $1,250,000 for step-up and step-down substations to connect the generating plant and the Minneapolis system with this high-voltage line. Fixed charges on this would be another $140,000. It is obvious that if we followed one of the popular conceptions of superpower, we would saddle on the shoulders of the present generation in our community about $300,000 per annum of unsound fixed charges. The conclusion that can be drawn from these figures is that we haul coal cheaper than we can transmit energy if the distance is much in excess of 50 miles.

At present the Northern States Power system covers a territory some 450 miles wide east and west and about 230 miles long. Many of these towns could operate a small modern efficient generating plant cheaper than we can serve them by an individual transmission line from any one of our power centers. However, these so-called transmission lines pass through countless small communities that could not have economical electric utility service

*Extract of paper published in the May Bulletin of the Minnesota Federation of Architectural and Engineering Societies.

†Vice-president, Wisconsin Power & Light Co.

were it not for the through lines that serve the larger communities.

As another illustration, there may be cited the power plant at Sioux Falls, South Dakota. Now the radical super-power element would undoubtedly recommend building a transmission line from Minneapolis to serve this territory. Since the steam plant at Sioux Falls is in fairly good shape, let it be assumed that it is kept for stand-by and peak-load service and a single-circuit, 250-mile, 110-kv. line be built to supply the normal power demands. This would cost about $2,000,000 and represent an annual charge of around $300,000 for fixed charges and maintenance. Now by good luck and extra good management the cost of generating might be cut by $150,000, leaving the present generation to shoulder the other $150,000 per annum.

Many years ago Samuel Insull made the statement that he could go to any place in the United States and build a modern super-steam plant and compete with any hydro development. In spite of popular conception he was right then, and is more than right today.

Hazards in Hydro-Electric Stations*

By C. A. Sears†

Less than one-third of all accidents are caused by faulty equipment, while the remaining two-thirds are caused by the failure of the human element. Small isolated plants which need a safety organization most, very often do the least along safety lines. Such companies are, however, being absorbed into larger systems with safer plants resulting.

A rigid clearance system is of first importance in the prevention of accidents in hydro-electric plants. Along with it there should be provided some safe method of giving orders by telephone. Wheelpit dangers are minimized by placing stop-logs ahead of the head-gates when men are at work in the wheel case.

Poor housekeeping is a conspicuous hazard in power-station work. Good housekeeping, on the other hand, spells safety. Lack of proper instructions is another conspicuous hazard and is largely responsible for the high percentage of accidents resulting from the failure of the human element.

Education is the important thing in safety. Personal supervision and inspection, training in first aid and resuscitation, newspaper articles, safety posters and slogans, rule books and executive letters all have their place and all contribute to the training of the mind of the individual instinctively to act safely—which is the object of safety education.

The expense of safety work that will accomplish results is a moral obligation of our companies and is as legitimate a part of the total cost of a manufactured article as the cost of the materials that are used.

*Abstract of a paper presented before the Fourteenth National Safety Congress, Cleveland, Ohio, Sept. 28 to Oct. 2, 1925.

†Manager, Mississippi River Power Co., Keokuk, Iowa.

Hazards in hydro-electric stations originate from two general causes: First, the material side—failure to provide proper mechanical safeguards for workmen engaged in hazardous work, faulty design of a building, or its equipment. Second, the human side—a failure on the part of a foreman to instruct employees in their duties, failure on the part of employees to think and act safely. I would venture a guess that about one-third of all accidents in industry can be prevented either by proper design of the equipment or the installation of suitable mechanical safeguards. For the elimination of the remaining two-thirds we must turn to the individual—we must educate him in ways of safety. We must kindle his co-operation in safety matters by a genuine and personal interest in his welfare.

The most serious hazard to guard against in a hydro-electric station is the danger of closing a switch when men are at work on the line and starting machinery while men are still at work. In a large station this requires special attention. In the first place it is necessary to place operating control in the hands of one person who is solely responsible for giving orders to close switches or to start any important equipment in the station. The next step is to provide a rigid clearance system which all men working on the system understand.

Danger of Flooding Wheelpit Another Serious Hazard

Another serious hazard which exists in almost every large hydro-electric station is the danger of flooding the wheelpit when men are at work inside. As a safeguard against failure of a headgate most companies have adopted the practice of placing stop-logs in front of the head gates as an added precaution. Whenever a man is at work on a main unit, some companies have adopted the practice of putting jacks in the operating cylinders of the gate-opening mechanism. In case of accidental opening of the valve or a failure in the piping system, it would then be impossible for the gates of the turbine to open and start the unit.

The next hazard in order of importance which is second only to that of the clearance system, is that of poor housekeeping. If there is anything about a hydro station which makes for carelessness, it is poor housekeeping. With an untidy station a man does not hesitate to add a little more confusion and dirt to that which already exists. Dirt and confusion mean accidents. With everything cleaned up and everything neat, an operator begins to take pride in his equipment and will pay attention to safety. Good housekeeping spells safety in most plants.

Instruction of Employees Often Neglected

The third hazard in a hydro-electric station in order of importance is the lack of proper instruction of employees in their duties and in matters of safety. Oftentimes a foreman is lax in instructing his men because he perhaps feels that the workmen know as much about their work as he does. To assume that the other man understands what you understand, especially in matters of safety, is to create a hazard—the hazard of ignorance.

Personal supervision and routine inspection of equipment is one of the best accident-prevention measures that can be instituted. One man out on the job looking for trouble or defects, for hazards and for poor housekeeping, will do a world of good in preventing accidents. This is one of the best safety measures we know of. While such work is not policing, it makes men think. It sets an example, and we know that one example of work well done is far more eloquent than words can possibly be. Another effective way of getting across the safety idea to employees, is training in first aid and resuscitation.

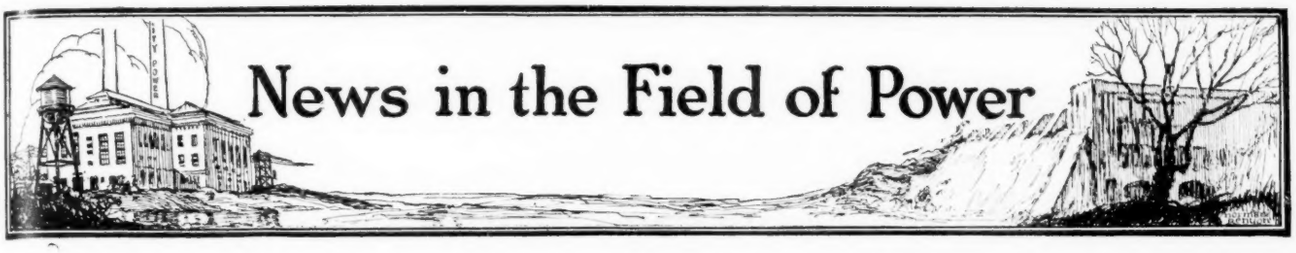
One more thought in regard to the cost of safety work. Just as in building a bridge, the designer is morally bound to make it a safe bridge, so in our business all of us are under moral obligation to provide a sufficient factor of safety to protect our employees and the public fully from the hazards that may be peculiar to our enterprise. No one questions the cost of providing a suitable factor of safety in the design of a bridge where human life is at stake. By the same token, no one should question the cost of providing for safety in our plants and throughout our organizations.

Project Planned for White River.—On the very day of the expiration of the legislative extension of the preliminary permit of the Dixie Power Co. a new application was filed by the White River Power Co., of Little Rock, Ark., of which Hugh L. Cooper is the consulting engineer. The new application provides for two dams in the White River and mentions the possibility of the need for six other dams in order to secure the maximum utilization of the river. It is estimated that equipment capable of generating 250,000 hp. can be installed to advantage.

The Dixie Power Co. held a preliminary permit covering this development which was extended to the limits provided in the water-power act. On the insistence of Senator Caraway, of Arkansas, legislation was approved granting an additional special extension of eighteen months. The Dixie Power Co. failed to apply for a license within the period.

The power is intended for use in St. Louis.

The State of Iowa has abandoned public ownership of its generating plant, maintained for twenty years to supply light and power for the State Capitol and surrounding grounds and public buildings. The state executive council, after an exhaustive study of costs in which it was assisted by the electrical engineering department of the Iowa State College, signed a contract with the Des Moines Electric Light Co. for the purchase of all its energy. It was found that the cost for service from the private company would be one-half the amount that is now expended. The minimum annual consumption is 750,000 kilowatt-hours.



Buffalo General Electric Co. Buys Well-Type Burners

The Fuller-Lehigh Co., Fullerton, Pa., has closed a contract with the Buffalo General Electric Co. covering well-type burning equipment, described on page 926, June 9, *Power*, for four B.&W. boilers of 12,515 sq.ft. of heating surface each. The coal-preparation equipment will consist of eight 46-in. Fuller air-separation-type pulverizers having a total capacity of about 60 tons per hour. This extension to the Black Rock station is being designed for continuous operation at slightly over 500 per cent normal boiler rating.

International Power Test Codes To Be Prepared

The International Electrochemical Commission, constituted by representatives of twenty-two countries, a number of years ago organized an "Advisory Committee on Prime Movers for Electrical Plant." While up to date this committee has not been very active in this field, it completed a report on "Nomenclature of Hydraulic Turbines," which was published by the I. E. C. in July, 1914, as Publication No. 29.

New impetus to activity in this field by the I. E. C. was supplied by a resolution drafted and indorsed by the delegates to the World Power Conference at Wembley, July, 1924. The import of this resolution was to request the International Electrochemical Commission to undertake in an energetic way the development of a series of codes or specifications for the testing of steam engines, steam turbines, hydraulic turbines as well as oil and gas engines. In requesting the I. E. C. to undertake this task, the representatives of the power-producing industry expressed the hope that the machinery established and the experience so far obtained in the development of international electrical standards may be put to good use in the formulation of a set of International Power Tests Codes.

Recent informal conferences on this subject indicate that when finally completed each of these international codes will be written in the form of an acceptance test of the prime mover in question and will consist of at least the following four sections: Nomenclature; instruments and apparatus, their types and limitations; important measurements, data and results, including some essential computation methods.

At the request of the United States National Committee of the I. E. C., The American Society of Mechanical Engineers has accepted membership on this national committee and has appointed Dr. Fred R. Low and Dr. William F. Durand as its two representatives on the Committee and C. Harold Berry as their alternate. The United States National Committee recognizes the society's long experience in the formation of power test codes and has cordially welcomed its offer to co-operate in this important and timely international undertaking.

Grand Falls Power Project Approved by New Regime

Developing of the Grand Falls power in New Brunswick, reputed to be the largest water power in eastern Canada, east of Niagara, will be proceeded with, according to a statement of the provincial premier in St. John. The late Veniot Government had given contracts on the eve of the election for the construction of the first unit, the cost involving several million dollars.

These contracts were held up by the new government immediately they assumed office, in order to have a further inquiry made into the merits of the scheme. It is the largest industrial undertaking which the province has yet experienced. It is situated above Fredericton, on the St. John River, much of the headwaters of which are in the State of Maine.

Novel Project Suggested for Holy Land

Among the many schemes for the development of Palestine one of the most original is that for utilizing the Dead Sea as a source of water power. As the Dead Sea is about 1,300 ft. below the ocean level and as the sun sucks up the River Jordan, which enters it, as rapidly as it runs in, the engineers calculate that if water were siphoned in from the Mediterranean, it could produce power and then be continuously evaporated from the expanded surface of the sea and the soaked sands of its shore. It is expected that this method will provide 450,000 hp. for the electrification of the Holy Land. The French Academy of Sciences, before which this scheme was presented, considered also the power possibilities of the other subsea sinks of the world, especially the Salton Sea, the Caspian Sea and certain sections of the Sahara.

The scheme would give an enlarged port at Haifa, entail the building of eight locks which would permit of navigation up to the Jordan Valley, a distance of over 40 miles. At an elevation of 869 ft. below sea level it is proposed to construct the first of the two hydro-electric stations, the available head being 1,125 ft. The plans include the damming of the Lake of Galilee and several streams and the use of the water so held up for irrigation.

New Bill Against Chicago's Diversion of Lake Water

Diversion of water from Lake Michigan by the Chicago Sanitary District entered a new phase of court procedure Oct. 5, when, on the opening day of the term, a motion was entered by Attorney General H. L. Ekern of Wisconsin for permission to file an amended bill in the suit of that state against the State of Illinois and the Sanitary District in which amended bill the States of Minnesota, Ohio and Pennsylvania join Wisconsin as complainants.

Attorneys representing Illinois and the Sanitary District, who were present, immediately filed notice of opposition to the motion for an amended bill.

The amended bill would replace the complaint filed by Wisconsin in 1922, on which no action has been taken before the Supreme Court because of priority for the suit of the Federal government against the Sanitary District to restrain excessive diversion, which was decided in favor of the government.

Anthracite Operators Accused of Mixing Sizes

John L. Lewis, International President of the United Mine Workers of America, in an address before the American Federation of Labor convention at Atlantic City on Oct. 14 charged, according to press reports, that the anthracite mine operators were muleting the public out of an annual rate of sixteen to twenty million dollars by a resizing plan they had forced upon retailers. He said in part:

"The anthracite operators are extorting from the public a sum which ranges between sixteen million and twenty million dollars annually to which they have no decent claim. When they appeal to the public for support, they come into court with unclean hands. In April of this year they put through a resizing deal with the retailers which gives the operators a chance to work off their steam sizes on the public by mixing them with the domestic sizes and charging the old prices. The retailers feared that if they protested they would be cut off and given no more coal to sell. The operators stand to make between sixteen and twenty million dollars annually by this deal to retain the old trade names and lower their quality. The public is now paying for the old names at the old prices and getting much smaller stuff to burn.

"They not only lowered the sizes outright, but introduced the round mesh in place of the square mesh in the breaker screens. This move alone means a reduction of one-fifth in size of practically every grade of coal."

Secretary Hoover Advocates State Regulation of Public Utilities

Analyzes the Objections To Local Control—O. C. Merrill Also Advocates State Regulation

IN A SPEECH delivered before the thirty-seventh annual convention of the National Association of Railroad and Utilities Commissioners at Washington, D. C., on Oct. 15, Herbert Hoover emphatically advocated the continuance of local and state control and regulation of electrical public utilities rather than of a centralized federal supervision. He said in part:

"I am proud of the progress in the theory and practice of utility regulation made in the last 30 years, a large proportion of it due to the state commissions. It has brought great economic values to the nation and an assurance of justice and protection of our citizens. I would not have that progress cease.

"In the practical state in which the nation finds itself today in the matter of power rates, three facts do not seem inapropos. The first is that regulation and progress in the industry is providing here a commodity—electricity—at pre-war prices. It is about the only one in the nation. The second fact is that the total distribution of interest and dividends by the electric companies taken as a whole has never yet exceeded 6 per cent upon the values of all the properties for rate-making purposes. The third fact is that the industry has expanded to meet public demand by fourfold in ten years. The state commissions may well claim a large portion of the credit for these conditions—their regulation has been effective.

"Before we abandon the system, let us be sure that we have something better to take its place.

"Until lately there has been no suggestion of federal regulation. The states have over a period of 30 years gradually developed the body of law and the administrative machinery which met the problem. No new question would arise today but for the great revolution in the organization of this industry. Because of this revolution four arguments are now advanced for substitution of federal for state regulation:

Arguments for Federal Control Analyzed

First, it is contended that watering of stocks is now in progress in the operating or holding companies, and that this inflated capitalization may be reflected in increased rates;

Second, that the extension of lines across state boundaries is creating a legal interstate problem over which the states have no authority;

Third, that some of the states have not yet set up adequate administration for protection of public interest and that this delinquency in state action warrants federal intervention;

Fourth; that this evolution of industry has developed "trusts" which will defy public interest until democracy will not be able to rule in its own house.

"The first argument that overcapitalization or watering of stocks places a burden upon the consumers falls to the ground at once if all the states universally adopt the principles of state regulation now demonstrated in many of them. There may be a watering of stocks in progress. But if rates be based either upon the true value of plant and equipment or upon actual investment, watered capital does not and cannot affect consumers.

"The second of the reasons advanced for the government taking over control is that owing to enlargement of production and distribution, electricity has become such a matter of interstate commerce as to place it outside the control of any one state. This requires analysis, for the question has both economic and legal phases.

"The argument is sometimes used that the power situation is parallel with the railroads, where federal regulation has been found necessary. It differs in several profound respects. Power has no such interstate implication as transportation. The grower of oranges shipping from California is affected by the freight rates of every different railway all the way to the Canadian border and the Atlantic seaboard. The same farmer as a buyer of power is interested in only one power company—the one in his immediate neighborhood. What the rates may be on 200 other power companies between him and the borders of the continent do not affect his rates in the remotest degree. Therefore one state is not vitally interested in the rates of another state as in the case of the railways.

"Furthermore, there has been outrageous exaggeration of the probable extent of interstate power. For economic reasons these power districts will in but few cases reach across state lines. There will be interconnection between systems of different states, but even including this the proportion of actual interstate movement in power to the intrastate movement will be comparatively small. At the present time less than two per cent of the power developed passes state lines. And if we except the special situation of one particular power plant there is only one per cent.

"But interstate problems arise only where the activities of the operating companies extend beyond state lines. They fall into two classes: First, where the same company is engaged in generation and distribution over a district embracing parts of two or more states, and second, where an operating company purchases power generated in a foreign state—the latter instance also embracing the 'interconnection' between districts lying in different states. As to the first class it has been well established by the courts in analogous instances that the state commissions have the power and authority to establish 'reasonable rates' to their consumers whether the property may be wholly within the state or not. These cases take care today of the large proportion of the 2 per cent passing our state lines. There remains, therefore, a small fraction of the problem, where the distributing company buys power by interconnection or otherwise from outside the state. Here the question is simply as to whether the rate paid for the purchased power is reasonable. If unreasonable, the commission may refuse to allow its full amount in settling the rate base. It is open to the commission to fix a rate for resale to consumers within its jurisdiction based upon what the commission considers a fair price, the companies having the usual recourse to the courts for redress against injustice.

"The third contention is that some of the states have not or will not set up adequate machinery for protection of public interest, and that, therefore, the federal government should intervene to protect their citizens. This is one of the oldest contentions for all sorts of federal action—that the nation shall intervene to remedy evils which the states can remedy but which some of them fail to remedy.

"The fourth contention is that this evolution of the industry has developed 'trusts' which do or will defy public interest. I have already stated that 'interconnection' in itself is no more evidence of a 'trust' than the exchange of railway cars. Also I may say that the ownership of 100,000 hp. would have been a 'trust' only fifteen years ago. To-day in order to get cheap power we build that much in a single machine and somebody has to own it.

"There is an economic question involved in this matter of the development of power 'districts' which I believe needs the attention of the state commissions, and action upon which would cure the evil if such should develop. There is a limit to the size to which operating 'districts' should be extended. The economies of central generation, of best diversity of load, have a territorial limitation in each case, beyond which no advantage is gained. Moreover the attention to service and response to local responsibility decreases after some maximum area has been reached. The best use of surplus power from one 'district' to another can be reached by purchase and sale through interconnection of 'districts.' Moreover it is important to preserve a large number of 'districts' in order to stimulate invention and improvement and, upon failure of a district operating company, to thus keep open the door of competition as the final remedy. Therefore, it seems to me highly desirable that the state commissions should study this problem in its broad aspects and through their undoubtedly existing powers scrutinize and disallow consolidations unless economic value to the commodity can be demonstrated."

Merrill for State Regulation

In speaking before the same convention, O. C. Merrill, executive secretary of the Federal Power Commission, after outlining the scope and supervision exercised by the commission over federal power projects arrived at the conclusion that state rather than

federal or municipal regulation is to be preferred. He also says in part:

"It is doubtless true that state regulation has not always met with success. That is not, however, the fault of the principle but of its application. There has been too little honest attempt to give it the opportunity to succeed. I have said elsewhere, and I repeat here, that if state regulation is to be successful it must be removed from the atmosphere of politics. The personnel of our public-service commissions should be of as high a character and ability as are the judges of the courts and should be as far removed from the influence of popular prejudice or the whims of executives or of legislatures. Then and only then will they possess the character as individuals and the stability as institutions necessary to cope successfully with the ever-increasing complexity of public-utility problems.

"As we view the field of public-service regulation, we observe at once that the majority of its activities fall within state lines. It is only in those instances where the energy delivered to the consumer is imported from without the state, or where power generated within the state is carried beyond its borders, that the transaction passes beyond the legal jurisdiction of the state. Such transactions constitute the limitations upon complete state control to which I have already referred."

Arch-Dam Investigations Making Progress

The arch dam investigation being conducted by the A.S.C.E. and sponsored by the Engineering Foundation, which includes the testing to destruction of a full-sized dam on Stevenson Creek, near Fresno, Calif., made further progress at a two-day conference at the Bureau of Standards on Sept. 28 and 29.

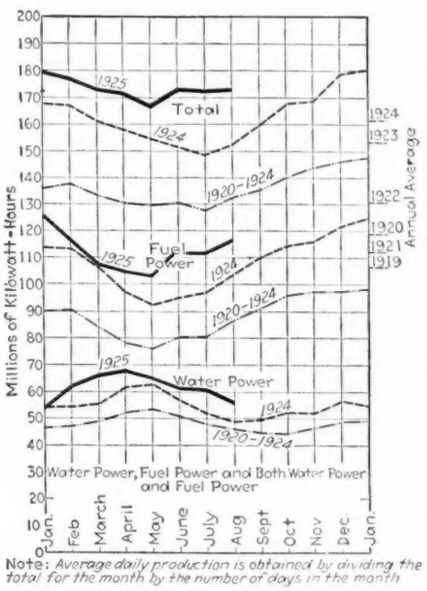
Special instruments and methods have been devised by the members of the Bureau of Standards staff. Col. E. Lester Jones, director, and Col. W. E. Bowie of the Coast and Geodetic Survey are also co-operating in suggestions of methods and equipment for exact measurements for determining the deflections or bending of the dam under pressure of water and other forces. Temperatures will be measured in the concrete of the dam by both electrical resistance coils, or thermocouples and thermometers. Excavations of the granite walls of the canyon for the foundation of the dam has begun, and the placing of the concrete and the setting of many of the istruments will be accomplished in December. It is expected that two years will be required to complete the full program.

Meanwhile, investigations of certain instruments and some of the properties of concrete, have been begun at California Institute of Technology by Prof. R. R. Martel, through the courtesy of President Robert A. Millikan. At the University of California, in the engineering laboratories, extensive and very careful tests to determine many physical properties of concrete are progressing under the guidance of Prof. Raymond E. Davis, through the courtesy of Dean C. Derleth, Jr., and President W. W. Campbell. Prof. C. Derleth, Jr., of the University of California, is chairman of the main committee, and H. Hawgood, of Los Angeles, chairman of the subcommittee on test dam.

Fred A. Noetzli, a consulting engineer, of Los Angeles, is secretary of the committee, and W. A. Slater, well-known expert in investigation of concrete and other structural materials, now at the Bureau of Standards and formerly of the University of Illinois, has recently been selected to have immediate charge of the experimental work on Stevenson Creek dam.

Public-Utility Electric-Output Curve Constant

The average daily production of electricity for each month in this year by public-utility power plants and the proportion produced by water power were as follows: January, 179,700,000 kw.-hr.—30 per cent; February, 177,900,000 kw.-hr.—35 per cent; March, 173,000,000 kw.-hr.—38 per cent; April, 171,700,000 kw.-hr.—39 per cent; May, 167,500,000 kw.-hr.—39 per cent; June, 173,400,000 kw.-hr.—35 per cent; July, 172,500 kw.-hr.—35 per cent; and August, 173,100,000 kw.-hr.—32 per cent.



Midsummer power curve flattens out

The average production of electricity during June, July and August was remarkably uniform and there was no marked decrease in July as in previous years.

Public-utility power plants in the United States consumed 3,338,956 net tons of coal in August, according to a report by the U. S. Geological Survey. This compares with 3,166,703 tons in July, as shown by revised figures. Fuel-oil consumption by utilities in August totaled 743,213 bbl., compared with 852,947 bbl. in July as shown by revised figures.

Spain Experimenting With Oil From Low-Grade Coal

United States government economists are watching with interest the efforts of the Spanish government to develop oil supplies from the low-grade coal in the provinces of Leon and Teruel.

A Spanish commission was recently appointed to conduct or to supervise tests on oil from low-grade coals in Spain and to test the product not only on naval vessels but by the military organizations. Spanish oil requirements are increasing and the government is doing its utmost to develop domestic supplies.

International Boundary Power Project Arouses Interest

Members of the International Joint Commission have completed hearings at International Falls on the application of the Fort Frances Pulp & Paper Co. for permission to construct dams and raise levels of certain boundary lakes.

The commissioners decided that the entire water-power project, one of the largest ever proposed affecting boundary waters, will be placed in the hands of United States engineers for study and that no decision on the application will be made until the engineers complete their report.

Plans for the international water-power project in the chain of lakes on the border between Minnesota and Ontario were placed before the commissioners at the hearing. The formal application of the Fort Frances Pulp & Paper Co. asked "the right to dam outlets of several lakes along the international waters above Rainy Lake, for the control of waters flowing into Namakan Lake, as well as, in cases where power is available, the right to develop such power and erect power plants."

At the conclusion of the hearing, Mr. Clark, chairman of the commission, said that further hearings will be held, and also that he was amazed at the intense interest which had developed over the matter.

"But as we began to dig into the case," he said, "we concluded that the interest was justified. It is one of the most interesting controversies that has been presented to the commission since I have been a member. On the one hand is a strong and well-established corporation, producing essential commodities, giving employment to hundreds of workmen and seeking to enlarge its facilities by expensive water-power development.

"On the other are the groups representing sportsmen and nature lovers who frequent the Canadian boundary country, backed by the summer-resort owners and citizens of the communities of the vicinity, who contend that development of the boundary river as a great source of power and as a headquarters of the logging industry, will ruin the lakes and waterways. The commission has its work cut out for it and the decision will be of prime importance. That is the real reason for the intensive study and in-

vestigation which the commission plans to devote to the subject before announcing any decision."

Chance for Young Draftsman with Government

The United States Civil Service Commission announces an open competitive examination for an apprentice draftsman for service in the departments at Washington. Receipts of applications close Nov. 10. Entrance salary $1,140, with chance of promotion up to $1,500 a year.

Testing Materials Society To Hold Important Meeting

The American Society for Testing Materials which is holding a three-day group committee session at the Hotel Cleveland, Cleveland, Ohio, Oct. 27-25 will have reports from subcommittees of interest to *Power* readers: A-5 Corrosion of Iron and Steel; C-8 Refractories; D-5 Coal and Coke; Joint Research Committee on Effect of Temperature on the Properties of Metals.

"The World Coal Situation"

In the *American Federationist* for October, Frank Hodges, secretary of the Miners' International Federation, writing on "The World Coal Situation," says: "The failure of international competition in coal is the logical outcome of the contraction of the coal market in the Old World, operating simultaneously with the expansion of coal production in the New World, this latter being represented by countries which up to the beginning of the last decade produced little or no coal at all, but whose requirements were met by importations from countries whose coal industry, was comparatively highly developed.

"South Africa, Australia, New Zealand, India, China and Japan were countries which until a few years ago absorbed European coal. Now they are producers of coal on their own account and what is more they are already in possession of small surpluses which on occasion push their way westward and northward in the teeth of the competition of the more highly developed countries of Europe and the North American Continent. He gives the following table:

	1913 Millions	1924 Millions	Increase or Reduction, Per Cent
Europe	605¾	549	— 9.1
America	533½	518	— 2.6
Africa	8¼	12	+45.2
Asia	55½	70¾	+27.8
Australasia	14½	18½	+27.6
Total	1,216¾	1,168¼	—27.7

"It is clear from the above that while Europe has declined in production by 56 million tons and America by 15 million tons, the continents in the Southern Hemisphere have increased production by 13 million tons. This change in the disposition of the world's coal trade will be continuous and progressive for the supplies of coal in the newer coal areas are practically unlimited.

WORLD PRODUCTION OF LIGNITE

	1913, Millions	1924, Millions	Increase or Reduction, Per Cent
Europe	126.9	163.2	....
America	0.02	2.1	....
Total	127.1	165.3	+30

"The world's shipping tonnage using coal in 1913 was 89 per cent. To-day the figure is 64¾ per cent. The following table is more expressive as indicating the increase in the use of oil fuel in the mercantile marine.

	1914, Million Tons	1924, Million Tons
Oil-burning ships	1¼	17
Tanks and motor ships	1½	7
Total iron and steel ships	42¾	37¼

"The fact that Great Britain, which has the highest surplus of coal available for export, should buy each year 5 million tons of oil mainly for fuel at an annual cost of £45,000,000 is one of the most paradoxical features of the present world coal crisis."

Practical Refrigerating Engineers Plans

The prospect for a large convention and educational exhibition of the National Association of Practical Refrigerating Engineers is becoming increasingly encouraging as the time for the convention approaches. The Detroit convention committee, composed of members of Detroit No. 1, with Charles W. Chapman, the national treasurer, as general chairman, are making arrangements for about 2,000, or twice the usual attendance. The convention and exhibition are to be held on the same floor, the second floor of the Statler Hotel, Detroit, Mich. The program committee has arranged for papers of value to the members and visitors and the refrigerating engineering profession in general. The date for this sixteenth annual convention and educational exhibition of the National Association of Practical Refrigerating Engineers is Dec. 8-12.

Stokers Sold During Past Nine Months

The Department of Commerce announces the following statistics on mechanical stokers for the months January to September, compiled from reports received from 13 establishments:

STOKERS SOLD AND KINDS OF INSTALLATION

Year and Month 1925	Establishments Reporting (Number)	Stokers Sold, Number	Installed Under: Fire-Tube Boilers — Number	Fire-Tube Boilers — Sq.Ft. Heating Surface	Water-Tube Boilers — Number	Water-Tube Boilers — Sq.Ft. Heating Surface
September	13	119	34	75,630	85	30,5920
August	13	91	27	38,800	64	259,850
July	13	147	30	50,750	117	536,440
June	13	128	26	46,250	102	394,700
May	13	122	23	39,340	99	516,540
April	13	120	23	58,930	97	417,340
March	13	131	17	24,460	114	686,530
February	13	135	28	49,050	107	413,930
January	13	57	4	8,700	53	270,010

A Correction

In an article on page 453 of the Sept. 22 issue of *Power* the statement was made in the article, "Maine Breaks Away from Isolation," that the legislatures of both Maine and New Brunswick have granted charters for the Passamoquoddy development. This should have read, the legislature of Maine has granted a charter and the authorities of New Brunswick have approved the development.

Personal Mention

Harold W. Merritt, formerly engineer with the Duplex Engine Governor Co., Brooklyn, N. Y., is now instructor in mechanical engineering, Cooper Union.

William D. Halsey, assistant chief engineer of the Hartford Steam Boiler Inspection & Insurance Co., Hartford, Conn., spoke on "Boiler Explosions, Their Cause and Prevention," at Swathmore College on Oct. 15.

Ernst Curt Loesche, Director der Curt von Grueber Maschinenbau Aktiengesellschaft, Berlin, Germany, and **Erich Schulz,** chief engineer, Municipal Light & Power Plant, Berlin, Germany, are visiting in this country studying pulverized-fuel installations.

William J. Ranton, of Rochester, N. Y., has just returned from an extensive trip including Winnipeg, Kansas City and Oklahoma, for the Fidelity & Casualty Co., with the Boiler Inspection Department of which he has been connected for many years.

Max Toltz has recently received the honorary degree of Doctor of Engineering conferred by the Ramsey Institute of Technology, St. Paul, Minn., in recognition of his high attainments in the consulting engineering profession and also because of his proficiency in the design of power plants.

Dean K. Chadbourne, who for five years has been connected with the Westinghouse Electric International Co. in the Far Eastern department, has been appointed general manager of this company, succeeding E. D. Kilburn, who has recently been elected vice-president of the Westinghouse Electric & Manufacturing Co.

Society Affairs

The Philadelphia Section of the A.S.M.E. will discuss "The Engineers in Wall Street," at its annual smoker at the Elks Club on Oct. 27.

The Lehigh Valley Section of the A.I.E.E. will hear Chester Lichtenberg of the G. E. Co. speak on "Automatic Stations and Their Remote Supervision," at its Oct. 30 meeting which will be held in the Physics Bldg., Lehigh University, Bethlehem, Pa.

The Mid-Continent Sections of the A.S.M.E. and A.I.M.E., Petroleum Divisions, will hear William G. Heltzel speak at the Oct. 23 meeting, which will be held at Tulsa, Okla. Dean Foster and S. A. Sulentic will discuss the subject for the A.S.M.E. and H. Wright and V. C. Millikan for the A.I.M.E.

The Utah Section of the A.S.M.E. will hold its Oct. 28 meeting at the Chamber of Commerce, Salt Lake City. "Economical Steam Generation," by Col. H. D. Savage, of New York City, will be the attraction.

The New York Section of the A.I.E.E. will hold its first meeting on Oct. 23, at the Engineering Building, 29 West 39th St., New York. Papers: "Three-Phase, 60,000 kva. Turbo-Alternators for Gennevilliers," by C. Roth, chief electrical engineer, Société-Alsacienne de Constructions Mechaniques, Belfort, France, and "Hydrogen as a Cooling Medium for Electrical Machinery," by Edgar Knowlton, Chester W. Rice and C. H. Freiburghouse, all of the General Electric Co. These papers have been published and discussion of them at the meeting is expected.

Business Notes

The Johns-Pratt Co., Hartford, Conn., has removed its New York City office from 41 East 42nd St. to 20 Vesey St., Room 610. This removal was made on October first.

The Brown Instrument Co., 4512 Wayne Ave., Philadelphia, Pa., announces the opening of a branch at 363 New High St., Los Angeles, Calif., with S. F. Godfrey, formerly of The Braun Corp., as district manager. A complete repair department will be maintained.

The American Schaeffer & Budenberg Corp., Berry and South Fifth Sts., Brooklyn, N. Y., announces that it has again established its New England office in Boston, at 80 Federal St. Coincident with this change the Boston plant will be transferred to Worcester, Mass.

The Lincoln Electric Co., Cleveland, Ohio, announces changes in its organization: J. M. Robinson, formerly manager of the Grand Rapids office, has been transferred to Detroit; G. W. First has been changed from Boston to Grand Rapids; J. E. Durstine, formerly of the Chicago office, is now at Buffalo.

The Liberty Electric Corp., Stamford, Conn., announces that Joseph McKay, Jr., on October first, became associated with the firm as the New York and Philadelphia district sales representative on Liberty valve operator and control equipment. Mr. McKay's headquarters will be at the offices of the company, 342 Madison Ave., New York City.

Trade Catalogs

Expansion Joints — Mogul Machine Co., Witherspoon Bldg., Philadelphia, Pa. Bulletin EXJ-1 describes double-end-guided expansion joints which fit all standard pipe sizes from 2 in. to 36 in.

Drills—Sullivan Machinery Co., 122 South Michigan Ave., Chicago, Ill. Catalog No. 80 describes the diamond core drills used for engineers test borings, coal and hard mineral prospecting, and heavy-duty types for oil-field drilling.

Coming Conventions

American Institute of Chemical Engineers—Dr. J. C. Olsen, Polytechnic Institute, Brooklyn, N. Y. Annual meeting at Sinton Hotel, Cincinnati, Ohio, Dec. 2-5.

American Institute of Electrical Engineers. F. L. Hutchinson, 29 West 39th St., New York City. Convention at Engineering Building, New York City, Feb. 8-12, 1926.

American Marine Exposition. Peter A. Sensenrg, 15 Park Row, New York City. Exposition to be held in the 212th Anti-Aircraft Armory, 62nd St. and Columbus Ave., New York City, Nov. 9-14.

American Society for Testing Materials. C. L. Warwick, 1315 Spruce St., Philadelphia, Pa. Group meeting at Hotel Cleveland, Cleveland, Ohio, Oct. 27-29.

American Refractories Institute. M. C. Booze, acting secretary, 2202 Oliver Bldg., Pittsburgh, Pa. Fall meeting at the Waldorf-Astoria Hotel, New York City, Oct. 29.

American Society of Mechanical Engineers—Calvin W. Rice, 29 West 39th St., New York City. Annual meeting at New York City, Nov. 30-Dec. 4.

American Society of Refrigerating Engineers, William H. Ross, 154 Nassau St., New York City. Annual meeting at New York City, Nov. 30-Dec. 2.

American Society of Safety Engineers. G. S. Wood, 29 West 39th St., New York City. Meeting at New York City, Nov. 18.

American Welding Society. M. M. Kelly, 33 W. 29th St., New York City. Fall meeting and Exposition of Welding Equipment and Accessories at the Massachusetts Institute of Technology, Cambridge, Mass., Oct. 21-23.

Association for the Advancement of Science. Burton E. Livingston, Smithsonian Institution, Washington, D. C. Annual meeting at Kansas City, Dec. 28, 1925, to Jan. 2, 1926.

Association of Edison Illuminating Companies. Preston S. Miller, 80th St. and East End Ave., New York City. Annual meeting at New Arlington Hotel, Hot Springs, Ark., Oct. 19-23.

Eastern Ice Manufacturers Association. W. H. Ross, 35 Warren St., New York City. Convention at Hotel Chelsea, Atlantic City, Nov. 23-25.

Electric Power Club, S. N. Clarkson, B. F. Keith Bldg., Cleveland, Ohio. Fall meeting at Briarcliff Manor, N. Y., Oct. 19-22.

Master Boiler Makers Association. Harry D. Vought, 26 Cortlandt St., New York City. Annual convention at the Statler Hotel, Buffalo, N. Y., May 25-26.

Midwestern Engineering Exposition, Inc. George E. Pfisterer, 53 West Jackson Boulevard, Chicago, Ill., general manager. Exposition and power show to be held in Chicago, Jan. 26-30, 1926.

National Association of Practical Refrigerating Engineers, Edward H. Fox, 5707 West Lake St., Chicago, Ill. Sixteenth annual convention and educational exhibition at Statler Hotel, Detroit, Mich., Dec. 8-12.

National Electric Light Association. M. H. Aylesworth, 29 West 39th St., New York City. Forty-ninth convention and manufacturers' exhibition, at Atlantic City, Young's Million Dollar Pier, May 17-21.

National Exposition of Power & Mechanical Engineering. Fred W. Payne and C. F. Roth, Managers, Grand Central Palace, New York City. Exposition at Grand Central Palace, Nov. 30-Dec. 5.

National Marine Engineers Beneficial Association. George A. Grubb, 313 Machinists Bldg., Washington, D. C. Convention at Washington, D. C., Jan. 18.

Society of Naval Architects and Marine Engineers. Daniel H. Cox, 29 West 39th St., New York City. Convention at the Engineering Building, 29 West 39th St., New York City, Nov. 12-13.

Thermometers, Indicating—The Foxboro Co., Inc., Foxboro, Mass. Catalog No. 148 describes, with many illustrations, these indicating thermometers for turbines, cooling towers and other and varied uses.

Equipment, Power Plant — The Schutte & Koerting Co., 12th and Thompson Sts., Philadelphia, Pa. Catalog describing the oil, gas, gasoline, byproduct and power-plant equipment manufactured by this company.

Speed Reducers—Foote Bros. Gear & Machine Co., Chicago, Ill. Catalog No. 26, devoted to IXL spur and worm gear reducers, is an elaborate eighty-page treatise on these two types of reducers, covering construction and applications, with considerable space devoted to gear formulas, the solution of practical gear problems and tables giving ratios, weights, prices and other physical data.

Fuel Prices

COAL

The following table shows the trend of the spot steam market in various coals, f.o.b. mines; *mine run* except Pittsburgh gas slack:

Bituminous, Net Tons	Market Quoting	Oct. 12 1925
Pool 1	New York	$2.75@$3.10
Smokeless	Boston	2.01
Clearfield	Boston	1.70@ 2.10
Somerset	Boston	1.85@ 2.30
Kanawha	Columbus	1.55@ 1.85
Hocking	Columbus	1.50@ 1.85
Pittsburgh	Pittsburgh	2.00@ 2.15
Pittsburgh gas slack	Pittsburgh	1.50@ 1.60
Franklin, Ill.	Chicago	2.25@ 2.50
Central, Ill.	Chicago	2.00@ 2.25
Ind. 4th Vein	Chicago	2.25@ 2.50
West Ky.	Louisville	1.10@ 1.50
S. E. Ky.	Louisville	1.40@ 1.65
Big Seam	Birmingham	1.60@ 2.00
Anthracite, Gross Tons		
Buckwheat No. 1.	New York	$2.50@$2.60
Buckwheat No. 1.	Philadelphia	2.50@ 2.60
Birdseye	New York	1.60

FUEL OIL

New York—Oct. 15, light oil, tank-car lots; 28@34 deg. Baumé, 4⅜c. per gal., 36@40 deg. 5c. per gal., f.o.b. Bayonne, N. J.

St. Louis—Oct. 7, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.85 per bbl.; 26@28 deg., $1.90 per bbl.; 28@30 deg., $1.95 per bbl.; 30@32 deg., $2.00 per bbl.; 32@36 deg., gas oil, 4.91c. per gal.; 38@40 deg., 6c. per gal.

Pittsburgh—Oct. 6, f.o.b. local refinery; 30@34 deg., fuel oil, 5¼c. per gal.; 36@40 deg., fuel oil, 5⅜c. per gal.

Dallas—Oct. 10, f.o.b. local refinery; 26@30 deg., $1.45 per bbl.

Philadelphia—Oct. 8, 27@30 deg., $2.15@$2.21 per bbl.; 18@22 deg., $1.848@$1.908; 13@19 deg., $1.575@$1.635 per bbl.

Boston—Oct. 13, tank-car lots, f.o.b. heavy oil, 12@14 deg. Baumé, 4.2c. per gal.; light oil, 28@32 deg. Baumé, 5½c. per gal.

Cincinnati—Oct. 12, tank-car lots, f.o.b. local refinery, 24@26 deg., Baumé, 6¼c. per gal.; 26@30 deg., 6¼c. per gal.; 30@32 deg., 6½c. per gal.

Chicago—Oct. 6, tank-car lots, f.o.b. Oklahoma, freight to Chicago 92c. per bbl.; 24@26 deg., $1.10 per bbl.; 30@32 deg., $1.25.

New Plant Construction

Ariz., Yuma—Bureau of Reclamation, E. Mead, Comr., will receive bids until Nov. 16 for the construction of the building and installation of machinery, including two 1,160 hp. vertical hydraulic turbines, generators, transformers, switchboard, etc., for siphon drop power plant, Yuma Project. Furnishing structural material and electrical equipment not included in this bid.

Calif., Fresno — San Joaquin Light & Power Co., Light & Power Bldg., is in the market for 2 water wheels, 1 on each end of generator shaft, total capacity 40,000 hp., 33,000 kva. horizontal generator, two 200 kw. exciters, 6 truck type oil switches, motor operated disconnect switches, transformers, etc., for Balch Creek power plant.

Calif., Long Beach—City has had plans prepared for improvements in sewer pump dist. No. 8 including pumping equipment, etc. Estimated cost $165,000. R. D. Van Alstine is city engineer, H. Paterson, sanitary engineer.

Calif., Los Angeles—Los Angeles County has had preliminary plans prepared for the construction of a 12 to 14 story hospital. Estimated cost $1,000,000. Allied Architects Assn., Citizens National Bank Bldg., is architect.

Calif., Los Angeles—Edwards & Wildey, Co., Black Bldg., is having plans prepared for the construction of a 12 story studio building including steam heating system at 7th and Lebanon Sts. Estimated cost $600,000. Walker & Eisen, 701 Great Republic Life Bldg., are architects.

Calif., Oakland—City plans an election soon to vote $500,000 bonds for the construction of a warehouse and cold storage plant. W. W. Harmon, is city engineer.

Calif., Oakland—United Income Properties Co., of San Francisco, c/o J. L. Stewart, 702 Market St., San Francisco, Archt., awarded contract for the construction of a 14 story hotel at San Pablo, Market and 28th Sts., here, to L. Cohn, 1 De Haro St., San Francisco. Estimated cost $2,000,000.

Calif., Pasadena—Pasadena Ice Co., is having plans prepared for the construction of a cold storage plant. Estimated cost $60,000. Hamm & Grant, Inc., 607 Ferguson Bldg., Los Angeles, are architects.

Calif., San Francisco—California-Mason Realty Co., c/o Weeks & Day, California Insurance Bldg., Archts., will soon receive bids for the construction of a 15 story hotel on Mason St. Estimated cost $2,500,000.

Calif., San Francisco—Shea & Shea, 454 Montgomery St., Archts., will soon receive bids for the construction of a 5 story addition to hospital for St. Marys Hospital, Hayes and Stanyan Sts. Estimated cost $800,000.

Calif., San Francisco—U. S. Government, Washington, D. C., plans the installation of a 15 ton ice machine and 150 kw. steam generator, also changing oil burner into power plant system in Letterman General Hospital, San Francisco Presidio here. Construction Quartermaster, Ft. Mason, San Francisco, is engineer.

Fla., Maitland—J. A. Brown, Mayor, will receive bids until Oct. 27 for the construction of a new waterworks system including pump house, 75 g.p.m. motor-driven deep well pump, 75,000 gal. tank on tower, etc. J. B. McCrary Engineering Corporation, Citizens & Savings Bank Bldg., Atlanta, Ga., is engineer.

Fla., Stuart—City plans an election Oct. 27 to vote $25,000 bonds for the construction of a municipal electric plant.

Ill., Chicago — B. K. Goodman, Pres. Produce Exchange Building Corp., 111 West Washington St., awarded contract for the construction of a 10 story bank and office building at Racine Ave. and 14th Place, to McLennan Construction Co., 307 North Michigan Ave.

Ill., Chicago—C. F. Henry, 7455 Greenview Ave., will receive bids until Oct. 25 for the construction of a 16 story apartment including refrigeration plant, steam heating system, 3 elevators, etc., at Delaware Place and Seneca St. Estimated cost $2,000,000.

Ill., Chicago—Hotel Lexington, 22nd St. and South Michigan Ave., takes bids about Oct. 15th for the construction of a 10 story addition to hotel including 2 passenger and 1 freight elevator. Estimated cost $750,000. Gallup & Joy, 646 North Michigan Ave., are architects.

Ill., Chicago—Hydestone Building Corp., 1118 South Michigan Ave., awarded contract for the construction of a 12 story hotel and theatre building at 5101-5125 Blackstone Ave., to John A. Lindstrum Co., 5153 North Clark St. Estimated cost $1,500,000.

Ill., Staunton—Mayor and City Council, will receive bids until Oct. 30 for the construction of a water supply system including pumping equipment, two 700 g.p.m., 100 ft. head motor driven centrifugal pumps with automatic starters, one 1,000 g.p.m., 125 ft. head, one 500 g.p.m., 125 ft. head, one 5,000 g.p.m., 40 ft. head motor driven centrifugal pumps with hand starters, 100,000 gal. elevated tank, filter plant, etc. W. A. Fuller Co., Railway Exchange Bldg., St. Louis, Mo., is engineer.

La., Kenner — H. A. Mentz, Hammond Engr., will receive bids until Oct. 26, for the construction of a waterworks system, including 2 complete units of oil pumping equipment, pumping station, 100,000 gal. tank, 300,000 gal. reservoir, etc. Estimated cost $75,000.

Mass., Milford—Milford Electric Light & Power Co., 241 Main St., will soon receive bids for the construction of addition to plant. Estimated cost $40,000. E. L. Rawson, 6 Beacon St., Boston, is architect.

Mass., Monroe—New England Power Co., 35 Harvard St., Worcester, awarded contract for the construction of a 9,000 kw. power station and 90 ft. dam on the Deerfield River, here, to Power Construction & Engineering Corporation, 35 Harvard St., Worcester.

Mich., Detroit — Williams & Coughlan, 801 Kresge Bldg., Archts., is receiving bids for the construction of a 9 story apartment and office building, 2 story church and 3 story recreation building including steam heating system at West Grand Blvd. and 12th St., for West Grand Blvd. M. E. Church. R. H. Bready, 1955 La Mothe Ave. Estimated cost $850,000.

Miss., Louisville—City is receiving bids for the construction of a 500,000 gal. reservoir and pumphouse. F. A. Livingston, 201 Masonic Bldg., is engineer.

Mo., Buffalo—City voted $40,000 bonds for improvements to waterworks including pumping equipment, tank on tower, etc. Russell & Axon, McDaniel Bldg., Springfield, are engineers.

Mo., Malden—I. M. Morris, Mayor and City Council, will receive bids until Oct. 29 for furnishing and installing two 300 g.p.m. motor-driven triplex pumps, uniflow engine driven a.c. generator, switchboard, boiler, heater, feed pumps, etc. F. L. Wilcox, Chemical Bldg., St. Louis, is engineer.

Neb., Ashton—G. L. Polski, Clk, will receive bids until Oct. 29 for the construction of municipal waterworks system including Sect. 6, single stroke deep well pump, 50 g.p.m., 28 crank r.p.m. Alt. A., 2 stoke deep well pump, 70 g.p.m., 30 crank r.p.m. Sect. 7 7½ hp. 3 ph., 220 v., 1,200 r.p.m., a.c. motor. Alt. A., 10 hp. 3 ph., 200 v., 1,200 r.p.m., a.c. motor, etc. Estimated cost $19,000. Henningson Engineering Co., National Bldg., Omaha, is engineer.

N. Y., New York—A. Brisbane, 238 William St., will soon receive bids for the construction of a 15 story hotel on West 54th St. Estimated cost $1,500,000. G. B. Post, 101 Park Ave., is architect. E. Roth, 119 West 40th St., is associate architect.

N. Y., New York—A. B. Clark Co., 15 Park Row, is in the market for two 40 hp. upright power boilers.

N. Y., New York—Cranleigh Hospital, 37 Bridge St., awarded contract for the construction of a 9 story hospital on East 90th St. to W. N. Smith, 101 Park Ave. Estimated cost $500,000.

N. Y., New York—Hensle Construction Co., c/o G. S. Kingsley, 145 West 41st St., Archt., will build a 14 story apartment on West 134th St. by day labor.

N. Y., New York—Phipps Estate, 787 5th Ave., is having plans prepared for the construction of a 16 story office building at 5th Ave. and 47th St. Cross & Cross, 385 Madison Ave., are architects and engineers.

N. Y., New York—A. J. Thomas, 15 East 47th St., Archt., is preparing plans for the construction of a 5 story apartment at Sheridan Ave. Estimated cost $500,000. Owner's name withheld.

New York—Two Hundred West One Hundred Six Street, Inc., c/o Sugarman & Berger, 345 Madison Ave., Archts., has had plans prepared for the construction of a 15 story apartment at 203 West 106th St. Estimated cost $600,000.

N. C., Gatesville—City plans the construction of an electric light plant including lighting system. Estimated cost $25,000.

N. C., North Wilkesboro — The Carola Lumber Co., is in the market for a 250 hp., 3 ph., 60 cycle, Corliss engine with 300 kw., a.c. generator, two 3 hp., three 5 hp., one 15 hp., one 20 hp., two 25 hp., one 30 hp., three 50 hp. and one 60 hp. motors.

N. C., Winston-Salem—R. J. Reynolds, Tobacco Co., has had plans prepared for the construction of additional steam electric power plant, including boiler room and turbine room to increase capacity of present plant. Estimated cost $350,000. J. E. Sirrine & Co., Greenville, S. C., are engineers. Contracts have been awarded for equipment.

O., Cleveland—Superior Hotel Co., c/o Morris & Weinberg, 3608 Euclid Ave., Archts., is having preliminary plans prepared for the construction of a 9 story hotel at 426 West Superior Ave. Estimated cost $500,000.

Okla., Ardmore—J. F. Anderson, 304 C Street, S. W., desires information on gas, oil engine and steam engine electric lighting sets for chicken ranch.

Okla., Bristow—City is having preliminary plans prepared for the construction of a new 1,000 hp. Diesel engine electric light plant. Estimated cost $100,000. Engineer not selected.

Okla., Buffalo—Blasdel & Brady, is having plans prepared for the construction of a 15 ton ice plant including 120 hp. Diesel oil engine and electric generator. Estimated cost $30,000. Company engineers in charge.

Okla., Hinton—City awarded contract for extensions to electric light plant to D. C. Wilson. $23,858.

Tex., Arlington—H. M. Moore, Mayor, is having plans prepared for the construction of a new electric light plant. Estimated cost $40,000. Municipal Engineering Co., Praetorian Bldg., Dallas, is engineer.

Tex., Asherton—City plans an election to vote $35,000 bonds for the construction of a municipal electric light plant. R. W. Taylor, Mgr.

Tex., Huntsville — City plans improvements to waterworks system including new motor, compressor, etc. Estimated cost $40,000.

Tex., Kemp—City plans an election soon to vote $100,000 bonds for the construction of a new waterworks system including pumping equipment, disposal plant, etc. Koch & Fowler, Central Bank Bldg., Dallas, are engineers.

Wis., Milwaukee — Milwaukee Hospital, 22nd and Cedar Sts., will soon receive bids for the construction of a 5 story addition to hospital, including boiler house. Estimated cost $250,000. Clas, Shepherd & Clas, 445 Milwaukee St., are architects.

Ont., Hamilton—T. W. Jutten, Mayor and Chn. Bd. of Control, will receive bids until Nov. 16 for furnishing and installing 5 vertical sewage pumps, 2 horizontal standby sewage pumps, gasoline motors, etc. in the Gage Ave. pumping station. Estimated cost $195,000. W. L. McFaul, City Hall, is engineer.

Ont., Toronto — University of Toronto, Queens Park, plans the construction of a 144 x 260 ft. artificial ice arena, including a steam plant, electric refrigeration plant, etc. Estimated cost $200,000. T. R. Landon, Confederation Life Bldg., is engineer.